BELT CONVEYORS
AND
BELT ELEVATORS

BY

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PREFACE

This is intended to be a practical book. It is not a mere restatement of what already appears in trade advertisements, nor does it contain descriptions of installations of conveying and elevating machinery. It aims rather to explain principles and the reasons for doing things.

The present volume describes Belt Conveyors and Belt Elevators. These machines are so generally useful and suit so many kinds of materials under so many operating conditions that they are used to illustrate some of the principles underlying the general subject of the design and use of conveying and elevating machinery and to serve as an introduction to that subject.

The business of designing, making and selling such machinery is hardly more than forty years old; for thirty years of that time the author has been active in it, at the drafting board, in the shop, and in the field supervising the erection and operation of the machinery. For thirteen years the author was chief engineer of one of the largest companies in the business, was responsible for the design of all kinds of elevating and conveying machinery, and acquired valuable experience in dealing with suggestions and complaints from users of the machinery and in co-operating with them in improvements in design and manufacture.

The aim has been to present that experience in such a form as to be useful to men who have material to handle and who want to know more of the "how" and "why" of conveying and elevating by belts than can be told in the catalogs and advertisements of manufacturers. The information given will be of use also to consulting engineers who have to advise in the selection of the proper machinery to do certain work, to engineers and draftsmen who design such machinery, and also to students in technical schools and colleges.

Much of the information published in this book has never appeared in print and for a great deal of it, the author is indebted to his friends, some of them business associates, some of them business competitors. To each of them he returns his sincere thanks.

West Chester, Penna. FREDERIC V. HETZEL.

June 3, 1922.

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BELT CONVEYORS AND BELT ELEVATORS

SECTION I.—BELT CONVEYORS

CHAPTER I

GENERAL DESCRIPTION OF COMPONENT PARTS

A belt conveyor consists of a moving endless belt which supports material and which by its motion carries the material from one place to another. The belt is driven by a pulley, and is supported on both runs, going and coming, by rollers or by a runway. The material may be put on the belt by hand, shovel, chute or some other means, and it is removed from the belt by discharging it over the end pulley or by deflecting it at some point along the run of the conveyor.

The elements of a belt conveyor are, therefore:

1. A belt to carry the material and transmit the pull.
2. Means to support the belt, usually rollers or pulleys.
3. Means to drive the belt, usually a pulley or a pair of pulleys.
4. (*a*) Accessories for maintaining belt tension, such as take-ups.
   (*b*) Accessories for loading the belt, such as a chute.
   (*c*) Accessories for discharging the material, such as a chute or a tripper.
   (*d*) Accessories for cleaning and protecting the belt, such as housings, decks, covers, cleaning brushes, etc.

The belt is a flexible jointless structure which runs quietly at any speed; it is not ordinarily harmed by the actual conveying of the material it carries. Since the material does not come into contact with the moving surfaces of pulleys and shafts in which there are friction losses, these losses are relatively small and the power required for the transfer of material is generally less than in other forms of conveyors. The belt with its rollers weighs less per foot of run than other types of conveyors doing the same or similar work, and hence frames, bridges and other supporting structures are relatively lighter and cheaper.

Belt conveyors are suited to the carrying of all sorts of material, wet or dry, from the lightest to the heaviest, and in any quantity. They have been known and used for over a hundred years, but the most rapid development in their design and use has occurred since 1893.

The Belt.—The belt must have a certain flexibility in order to wrap around the pulleys, width enough to carry the required quantity of material and strength enough to bear the weight of the load and transmit the pull in the conveyor. These conditions can be met by bands of metal, leather, or woven fabric. For metal belts see page 60. Leather belts are expensive and do not resist wet and abrasion well enough in conveyors and elevators to justify their greater cost.

Belts of hemp fiber are used to some extent in Europe, but in this country practically all conveyor and elevator belts are made of cotton fiber. They are of several forms:

1. Rubber belts are made of layers or plies of cotton duck cemented together by an elastic rubber compound. In "friction surface" belts the outside of the belt is covered by the thin layer of compound adhering to the outer plies; in rubber-covered belts an extra layer of rubber compound is attached to the outer plies beyond the thin coating of "friction rubber." No attempt is made to waterproof the individual cotton fibers, the layers of rubber being depended upon to keep moisture out of the belt. See page 20.

2. Stitched canvas belts are made of layers or plies of cotton duck folded together to give the required width and thickness, and then sewed through and through with strong cotton twine. To waterproof the fibers and to reduce internal wear, the made-up belt is impregnated with a mixture of oil and gum. See page 46.

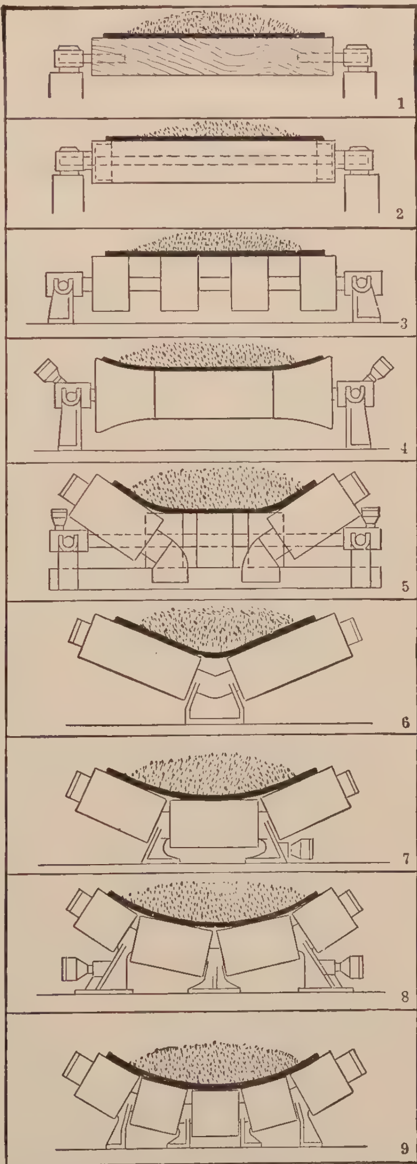
3. Balata belts are made of duck with the fibers of the cotton waterproofed by impregnation with a liquid solution of balata, a tree-gum similar in some ways to rubber. The impregnated duck is folded and rolled under pressure to make a belt of the required width and thickness, the balata gum acting as a cement to hold the plies together. See page 49.

4. Solid-woven belts consist of a number of layers of warp (lengthwise) threads and weft or filler (crosswise) threads woven and interbound together in a loom to make a structure of fabric of the necessary width and thickness. Most of them are waterproofed like stitched canvas belts, but some are impregnated with a rubber solution and then covered with a rubber sheathing. See page 50.

Supports for the Belt.—Supporting idlers consist of rollers or pulleys, of wood, cast iron or steel in various forms and combinations. For light work, especially in handling packages, the roller may be a cylinder of hard wood (Fig. 1), or a piece of thin steel tubing with inserted heads of wood or metal (Fig. 2). For heavier work, the roller may consist of several cast-iron pulleys mounted on a through shaft (Fig. 3), or the outer pulleys of the combination may be enlarged at their outside ends (flared idlers) in order to lift the edges of the belt slightly and prevent spilling (Fig. 4). When it is desired to increase the carrying capacity of a belt it may be bent into trough form by turning up the edges of the belt by troughing or "concentrator" pulleys (Fig. 5), or by supporting the whole width of the belt on idlers in which two, three, four or five pulleys are set at various

angles from the horizontal to trough the cross section of the belt (Figs. 6, 7, 8, 9).

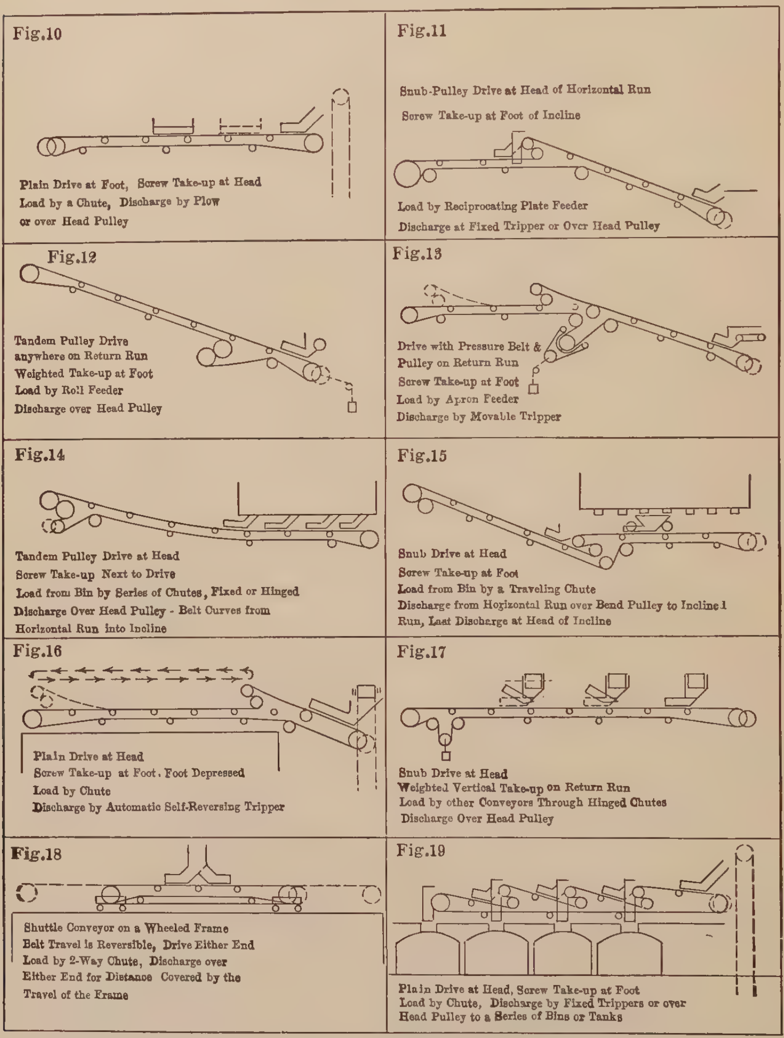
Driving the Belt.—The drive for a belt conveyor consists of one or two pulleys around which the belt wraps, suitably mounted on shafts and bearings and driven from a source of power through belts, chains, gears, or other means of power transmission. The simplest drive is that in which the belt wraps half way round the end pulley; this is usually at the head end or delivery end of the conveyor toward which the material moves, but it may be at the foot or loading end of the conveyor (Fig. 10). If 180° of belt wrap is not enough to drive the conveyor, it may be necessary to get a greater wrap on the driving pulley by the use of a snub or reverse-bend pulley (Fig. 11). For still greater driving contact, the belt may be led around two pulleys, both of which are drivers. Figs. 12 and 14 show two arrangements of these tandem-drive pulleys. To accomplish the same result, a pressure-belt (Fig. 13) may be used to increase the grip between a conveyor belt and a single drive pulley.



Figs. 1–9.—Supporting Idlers for Loaded Run of Conveyor Belts.

Belt-conveyor Accessories.—Conveyor belts in service stretch more or less, and it is necessary to have means to "take up" or remove the slack as it is formed. At some point in the conveyor, a pulley is mounted on a shaft running in bearings which are adjustable in position either by a screw or by a weight. Fig. 10 shows a conveyor with a screw take-up at the head end; in this case the tension required to pull the loaded side of the belt is transmitted through the take-up; the usual position for the take-up is at the foot end as in Fig. 11,

where the belt is under less tension. In Fig. 14 the take-up is at the point of *least* tension, that is, where the belt runs off from the driver. Fig. 12 shows a take-up shaft with a pull-back weight; in Fig. 17 a "gravity"



FIGS. 10–19.—Typical Belt Conveyors with Various Arrangements of Drive, Feed, Discharge and Take-up.

take-up on the return belt takes care of the slack and maintains driving contact on the end pulley.

Loading.—Loading chutes may be used in connection with a gate controlled by hand, as in drawing grain from bins, or to deliver to the belt a

supply of material at a uniform rate as in Figs. 10 and 14. If the material is hard to control by a gate, or if the supply is intermittent or irregular, feeders of various kinds (Figs. 11, 12, 13) are used in connection with loading chutes.

Discharging.—The simplest discharge is over the end pulley (Figs. 12, 17); sometimes a chute may be required there, often it is not. If the discharge is to be at some point short of the end of the conveyor, the material may be deflected sideways from the conveyor by a plow or scraper set diagonally across the belt (Fig. 10); more frequently this is done by inverting the belt or running it in S form through a tripper (Figs. 11, 12, 16, 19). In the tripper or " throw-off," as it is called in England, the material leaves the belt as it reaches the top of the upper pulley and is caught in a chute which directs it to one side, or by means of a by-pass gate, back on the belt again if the material is to be carried past the tripper as in Figs. 11 and 19. Trippers may be fixed (Fig. 10), or movable (Fig. 13), or traveling and self-reversing (Fig. 16), and may be operated by hand or power.

Protecting the Belt.—The belt is the most expensive part of a belt conveyor, often costing more than all the rest of the machinery and accessories combined. At the same time it is the most vulnerable part; it is subject to abrasion from impact of the material and to a number of injuries from negligence or accident. Injury at the loading point can be avoided by proper design of the chute; other mishaps can be prevented by care in operation and maintenance. To lessen the risk of certain other injuries it is necessary to provide means toclean the belt from adhering particles, to cover it where it is exposed to the weather, and to prevent it from being cut by objects falling against it or upon it.

CHAPTER II

DEVELOPMENT OF BELT CONVEYORS

History in the United States.—The earliest reference to the use of belt conveyors in American practice is in Oliver Evans' " Miller's Guide " published in Philadelphia in 1795. This describes and illustrates a flat belt receiving material on its upper run and discharging over the end; Evans calls it "a broad endless strap of thin pliant leather or canvas revolving over two pulleys in a case or trough." In the flour and grist mills built by Evans and his successors some of these conveyors were probably used, but it was more common in those days to use screw conveyors to convey grain as well as the lighter mill products.

Belts sliding in troughs (Fig. 20) were used before 1840 to convey materials which were not suitable for screw conveyors, such as clay, shavings, saw-mill

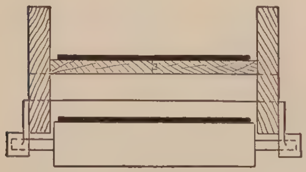


Fig. 20.—Conveyor Belt Sliding in Wood Trough, 1830.

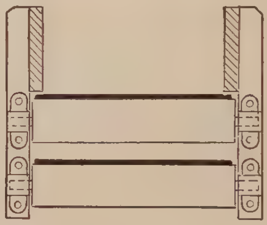


Fig. 21.—Clay Conveyor Belt Supported on Rollers, 1870.

refuse and crushed stone. Sometimes the return run slid back, sometimes it was carried on rollers. In some cases the sides of the trough were made with hinged sections which swung inward to stand diagonally across the belt and thus discharge material along the run of the conveyor.

When such conveyors were used for hard and gritty substances, the belt did not last long. In clay conveyors, the clay stuck to the sides and bottom of the trough, hardened there, and wore out the belt rapidly. Some of these troubles were avoided in the design shown in Fig. 21, which represents a form of clay conveyor in extensive use forty years ago. This construction substituted rolling friction in lubricated bearings for sliding friction on dirty rough surfaces; it saved power and helped to preserve the belt, although the material could not be carried without some leakage under the skirt-boards or trough sides, and the edges of the belt were still subject to injurious wear.

The great increase in the quantity of grain handled in this country after

1850 and the development of the grain "elevator" or storage system created a demand for belts of larger carrying capacity. Merrick & Sons, of the Southwark Foundry in Philadelphia, who built the Washington Avenue Grain Elevator there between 1859 and 1863, used wide composite belts which consisted of two parallel leather belts to which at intervals were riveted bent iron bars or spreaders to support a trough of canvas which carried the grain. This construction was similar to Fig. 22; power was applied to the edge belts by iron pulleys separated by the width of the canvas trough, or by a wooden drum grooved to clear the sag of the canvas. Narrow pulleys for idlers supported the edge belts on the loaded run and on the empty run.

Grain conveyors of this kind were installed in other American "Elevators" during the sixties. As used at Duluth early in the seventies (T. W. Hugo, Transactions A. S. M. E., 1884), they consisted of rubber belts 7 inches wide (Fig. 22) supporting a canvas trough 2 feet wide, which sagged 4 or 5 inches in the middle. They ran at 650 feet per minute and carried 12,000 bushels of wheat per hour. A composite belt of this kind had only 12 or 14 inches of belt width to engage the face of the driving pulley and hence the length of the conveyor was limited to what that width of pulley face would drive, but on the other hand, the load to be pulled corresponded to that of a belt 24 inches or more in width. Aside from that disadvantage the edge belts would not stretch alike, the spreader-bars would tear loose from their fastenings, and when the bars pulled loose, the belts came off the idlers and there was serious trouble.

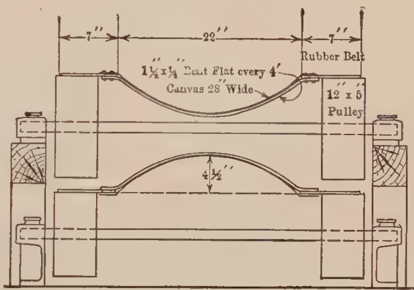


Fig. 22.—Composite Belt for Grain Conveyor, 1870.

Invention of the Tripper.—On February 10th, 1863, United States patent 37615 was issued to Oren C. Dodge of New York with the following claims:

1. "Delivering the grain at any desired point along the line of a traveling belt by bending such belt substantially as specified, for the introduction of a hopper or chute.

2. "A traveling belt for conveying grain, provided with vertical or nearly vertical edges, forming a trough."

Dodge's drawings show an arrangement of fixed trippers and a belt with low upstanding flanges and supported on cylindrical rollers.

Grain Conveyors at Liverpool.—About 1865, P. G. B. Westmacott and G. F. Lyster, engineers for the Birkenhead and Waterloo Docks at Liverpool, England, made experiments with a 12-inch belt which showed that belt conveyors carried more grain with less power than screw conveyors. In 1866 they installed a system consisting of a skip hoist for elevating the grain with belts for distributing it. The belts were 18-inch 2-ply rubber, run at

450 to 500 feet per minute and supported every 6 feet on straight wooden rolls. Discharge was effected by running the belt through a traveling tripper pushed by hand. Tuck-up or concentrator rolls mounted on portable frames were used to keep the grain back from the edge of the belt at the loading point and where the belt began to lift into the tripper. How much Westmacott and Lyster knew about American practice at that time we cannot say. Westmacott's account of their work (Proceedings Inst. of Mech. Engrs., 1869) makes no reference to the prior use of belts for grain in the United States nor to the use of trippers there. The traveling tripper was covered by Westmacott's British patent No. 3061 of 1866, and U. S. patent No. 66759, in 1867; it was new and was a great improvement over the fixed trippers shown in the Dodge patent.

Development of Grain Conveying.—The improvements made at Liverpool were taken up by American engineers, at first by W. B. Reaney in alterations at the Washington Avenue Elevator in 1873 and in the design of a complete new Elevator (Canton No. 1) for the Northern Central Railroad at Baltimore in 1876. In this Elevator, designed by Reaney and built by John T. Moulton & Son of Chicago, the conveyor belts were 30 inches wide 4-ply rubber and ran at 550 feet per minute over straight wood rolls placed every 5 feet on the carry and 10 feet on the return. Wooden concentrator pulleys similar to Fig. 23 were used at the loading points only. This work by Reaney introduced wide rubber belts into the business of handling grain. For a reference to a rubber belt which he installed at Philadelphia in 1873, see page 30.

The Canton No. 1 Elevator had the first traveling trippers in which the moving power was taken from the conveyor belt.

In other grain conveyors built in this country up to 1885, the belts were made to give greater capacity by troughing them over spool-shaped idlers (Fig. 23). As long as the difference between the middle and end diameters was only an inch or two, the pulley side of the belt did not suffer appreciably from the fact that the edge of the idler traveled more than the middle in feet per minute and hence must rub the belt. When the trough was made deeper by making the sag of the belt 2 or $2\frac{1}{2}$ inches (see T. W. Hugo's paper), the wear on the belt was great enough to be noticed; but the main objection to the deep spool idler as used on some grain conveyors was that a belt lightly loaded or empty was apt to run crooked. This trouble always exists where all or a great part of the weight of belt and load is carried on revolving conical surfaces or on separate cylindrical surfaces which are set to trough the belt, (see page 77). It led first to the use of guide pulleys to bear against the edge of the belt, always a dangerous and destruc-

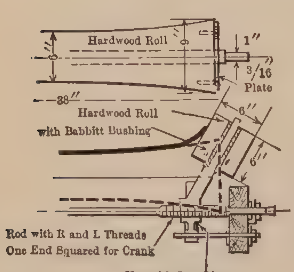


Fig. 23.—Carrying Idler and Concentrator Roll 36-inch Grain Belt. (John T. Moulton & Son, 1880.)

tive thing; then to the use of the "dish-pan" idler which came into use in the late seventies (Fig. 24). By having the end rolls or "dish-pan" loose on the shaft and free to turn independently of the horizontal pulley, there was less chafing of the belt than with deep spool idlers as long as the belt had a slight contact with the end rolls (Fig. 24, left half). Under a heavy load, however, the belt assumes the shape shown in Fig. 24, right half, and the pulley side is rubbed so as to cause a noticeable loss of power in driving the conveyor. Besides that, there was difficulty in making the belt run straight in the narrower widths, and after 1890 this form of idler gradually became obsolete.

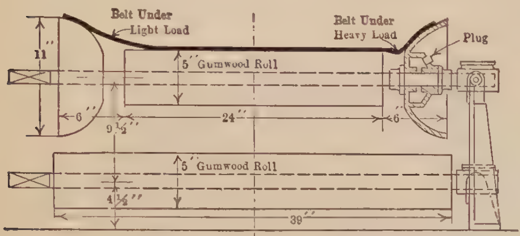


FIG. 24.— "Dish-pan" Idler and Return Idler for 36-inch Grain Belt, 1880.

After twenty years of experience with various kinds of belt idlers, designers of grain-elevator equipment in this country reached the conclusion that the right way to convey grain is by a belt that is flat (Fig. 25) or

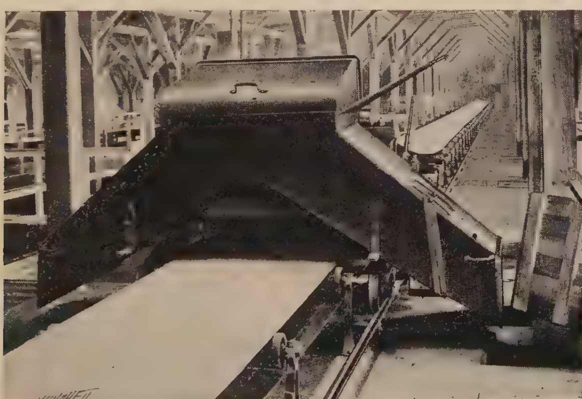


FIG. 25.—40-inch Flat Grain Belt, 1897. (Link-Belt Company.)

nearly so. A flat belt has a good contact with the horizontal idler pulleys and will run straight, empty or loaded, or even loaded to one side of the center. Grain carefully fed to a flat belt will not tend to shift toward the edges and spill (see Fig. 140), but to prevent scatter at the loading points and to increase the carrying capacity moderately it is customary to use inclined concentrator pulleys at the loading points only, or at intervals along the length of the conveyor (Fig. 26). The first concentrators used in this country were inclined at 60° (Fig. 23), but as many cases of longitudinal cracking of belts could be proved against the severe bend in the belt,

the angle was reduced to 45°, and then to 35°, which is now most common. Concentrators with angles of 30° and 22° are also made, to reduce still further the flexure of the edges of the belt.

In modern practice the concentrators may be used on a separate stand

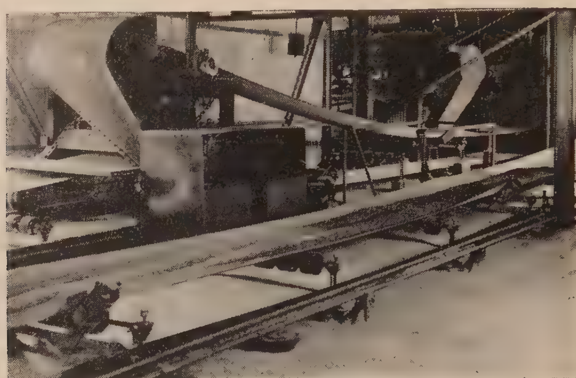


FIG. 26.—36-inch Grain Belts with Concentrators every 15 Feet. (James Stewart & Co., 1908.)

(Fig. 27) or combined with the horizontal pulleys (Fig. 28), or a stand may be used to take the return idlers also.

Conveying Materials Heavier than Grain.—With the development of conveying machinery between 1880 and 1890 belts were used for handling

FIG. 27.—Concentrator Pulleys Mounted on a Separate Stand.

coal, ore and other materials heavier than grain. Designers followed grain elevator practice as to belts and idlers. One of the largest installations in the early nineties was the ore concentrating plant of the New Jersey and Pennsylvania Concentrating Co. at Edison, N. J. It had over 50 belt conveyors, ranging from 20 inches to 30 inches wide and up to 500 feet long. The

first idlers used there about 1891 were heavy cast-iron spool idlers, but these were failures because in order to get the greatest capacity from the belts, the troughing was deep and the spools were much larger at the ends than at the middle. These were heavier than the spools used on grain conveyors; the work was continuous, the place dusty, and the belts wore

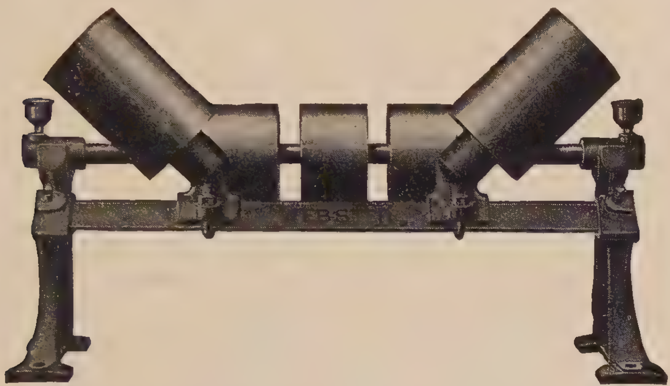


Fig. 28.—Concentrators Mounted on a Stand with Carrying Pulleys.

out rapidly. When the plant was rebuilt in 1893 idlers of the type shown in Fig. 29 replaced the spool idlers; the new idlers were practically the same as the flat rolls with concentrators used on grain conveyors.

The chief trouble at Edison and at other places doing similar work was with the belts. The sharp pieces of heavy ore cut the fabric and the stitching of canvas belts and loosened the plies; the same thing happened with rubber

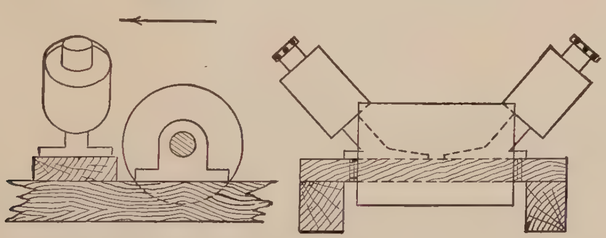


Fig. 29.—Troughing Idler for Ore Conveyor. (Edison, N. J., 1893.)

belts. We know now that much of the difficulty was due to improper delivery of material to the belts and to improper construction of the belts themselves. Fig. 30 shows a transfer between two belts at Edison; apparently no attempt was made to deliver material to the receiving belt with some velocity in the direction of travel.

Improvements by Thomas Robins.—Thomas Robins, Jr., then in the business of making rubber belts, visited the plant about 1891 and noticed the following points: 1, that the thin layer of rubber which covered the belt

resisted abrasion much longer than did the duck which formed the body of the belt; 2, that each layer of duck wore out faster than the one preceding it, showing that as the belt wore thin and the tension on the threads increased they were cut more easily; 3, that belts wore out in the middle and split longitudinally while the edges were still good (Trans. A. I. M. E., April, 1896). From these observations Mr. Robins concluded that the sole function of the fabric in a rubber belt should be to give the belt tensile strength and that it should be protected from injury by a cover of rubber compound that would resist abrasion better than the cotton threads or the thin layer

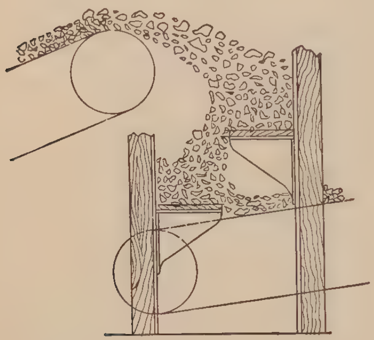


FIG. 30.—Transfer of Ore between Two Belts. (Edison, N. J., 1893.)

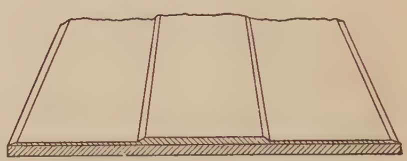


FIG. 31.—Original Robins' Belt with Extra Thickness of Rubber in Center, 1893.

of friction rubber that covered them. After experiments with various rubber compounds, he furnished belts with a thick face of rubber that proved to be much more durable than those previously used, lasting years where others lasted months. In 1893 he patented the belt shown in Fig. 31, which has an extra thickness of rubber at the center to resist the abrasion which comes from feeding material from a comparatively narrow chute; it resisted abrasion very well but the belt was so stiff that it would not conform to the shape of the troughing idlers, which, as shown in Fig. 29, had the side pulleys inclined at 45°.

In 1896 Mr. Robins patented his "stepped-ply" belt (Fig. 32) in which

FIG. 32.—Robins' Stepped-ply Belt, 1896.

the belt thickness is the same for the entire width but the plies of fabric are fewer at the middle than at the edges. The space left by the omission of the plies was filled with the rubber of the top cover, making the extra thickness of protective cover there equivalent to that shown in Fig. 31, but using less rubber where the abrasion was not so great. The omission of the plies at the middle made the belt more flexible in cross section, allowed it to

conform more closely to the shape of the troughing idlers and at the same time left a thick stiff edge to bear against the side guide pulleys (P', Fig. 33). These pulleys were necessary to keep the belt straight on 45° troughing idlers, especially when the belts were comparatively narrow and stiff and when the inclined idlers came every 4 or 5 feet. Idlers inclined at 45° were in successful use on grain conveyors, but there the belts were relatively wide and thin and the troughing occurred only at the loading points or at intervals not closer than 8 or 10 feet. Under these conditions the grain belt had a good guiding contact with the horizontal pulleys and did not usually require side guide idlers to keep it straight; but with stiff and narrow belts on 45° idlers, the guiding contact on the horizontal pulleys was slight, the side guide pulleys were necessary and the belt had to have a thick stiff edge; otherwise the edges were chafed, bent over or rubbed off and the belt destroyed.

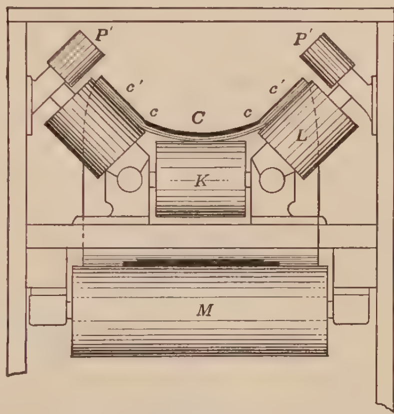


FIG. 33.—Stepped-ply Belt on 45° Troughing Idlers with Side-guide Pulleys, 1896.

In the early days of the belt-conveyor business, say before 1900, comparatively few belts wider than 24 inches were used. Troughing was considered a means to permit a narrow belt to carry what would otherwise require a wider and more costly belt, and the tendency was to use belts narrower than would now be considered good practice.

The Robins patent (571604, Nov. 17, 1896) covers also the idler construction shown in Fig. 34, with the following claims: "Claim 5. The supporting pulleys L, K, L, the hollow bearings F, therefor, and the horizontal and turn-up hollow-shafts secured in the said bearings, and the oil devices mounted on the ends of the turn-up shafts, substantially as set forth." "Claim 6. In combination the two brackets or castings suitably supported, the horizontal pulley mounted between them, the turn-up shafts secured in the said brackets or castings and the pulleys L loosely turning thereon, substantially as set forth." This idler followed earlier designs in having the troughing pulleys inclined at 45° (Trans. A. I. M. E., April, 1896), but it differed from them in combining the troughing idler group into one self-contained unit by using only two castings to support the pulley shafts instead of four (compare Figs. 34 and 29). This arrangement

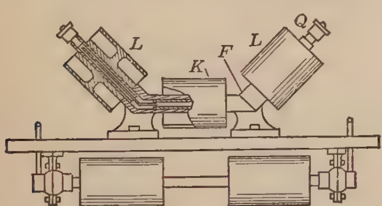


FIG. 34.—Robins' Three-pulley Idler, 1896.

with pulleys turning loose on hollow shafts permitted lubrication from two points instead of four, and at the same time the idler itself was easier to install and to use; there was no distinction between front and back as in the older form where the belt ran on to the horizontal pulley first (see Fig. 29). Although the figure in the patent shows the three pulleys in a single plane, that feature is not referred to in the language or the claims of the patent; it seems to have been adopted as a means of effecting lubrication in a certain way.

Prior to 1890 much had been done toward the improvement and standardization of grain conveyors by John T. Moulton, James A. Macdonald, the Webster & Comstock Mfg. Co. and others interested in the grain elevator business, but belt conveyors had not been used extensively for materials other than grain. Mr. Robins' great contribution to the industry was an active company which began in 1896 to design and sell belt conveyors for handling coal, ores, stone and all similar materials. Before that time most of the business in this line lay between the belt maker and the pulley maker, neither interested in the other's product and neither capable of giving engineering advice nor of insuring definite results from the operation of the conveyor. There was a great expansion in the use of all kinds of conveying machinery between 1895 and 1910 and the belt conveyor had its share of the growth. To Mr. Robins and his company must be given the credit for most of the pioneer work in extending the use of belt conveyors to the handling of materials other than grain and in laying the foundations for the engineering knowledge of the business. Others entered the field, some with devices intended to evade the Robins patents, others with improvements suggested by the growing experience of makers and users of belt conveyors.

Changes in Belt and Idler Construction.—The earliest Robins idlers had a rather wide gap beween the adjacent edges of the pulleys, a distance sufficient to allow a heavily loaded belt or an old limber belt to sag between the pulleys, as shown in Fig. 35. The same defect existed in various two-pulley (Fig. 6), four-pulley (Fig. 8) and two-plane three-pulley idlers, in which the edges were not brought close together. After the first few years of experience, the gap was made less, and in the majority of modern idlers, the distance between the pulleys is made as small as practicable, i.e., about $\frac{1}{4}$ or $\frac{1}{8}$ inch.

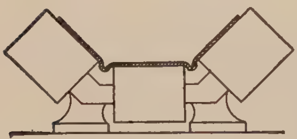


Fig. 35.—Sag of Belt into Gap between Idler Pulleys.

Two-pulley and four-pulley troughing idlers are now practically obsolete.

When the belt sags between the pulleys of a troughing idler of the kind shown in Fig. 35, it is not only in danger of cracking by direct flexure under the load, but there is a tendency of the edges of the pulleys to seize the sag of the belt and squeeze it together to form a sharp bend. Comparing Figs. 36 and 37, it is evident that the greater the angle at which the pulleys are inclined to each other, the greater is the wedge angle toward which the belt travels as it approaches the idler, and the greater the chance that the

converging edges of the pulley rims will seize the sag of belt and squeeze it. This squeezing action causes the belt to crack or to split lengthwise.

From examinations of spoiled and discarded belts, it was noticed that splitting was more apt to occur if the longitudinal joints or seams in one or more plies of duck came just at the point where the belt was bent by the troughing pulleys, or if that line of flexure in a stepped-ply belt happened to come where the thickness of the plies changed, as at *c*, Fig. 33. In early days belt makers were not always particular as to how the duck was cut and assembled. If the width was made up of several narrow strips of duck and

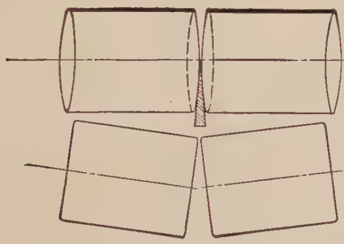


Fig. 36.

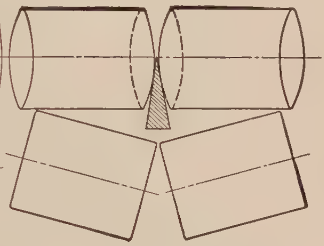


Fig. 37.

Fig. 36.—Angle between Pulleys 15° Small Wedge-angle.
Fig. 37.—Angle between Pulleys 30° Large Wedge Angle and Greater Liability to Pinch the Belt.

if the butt-joints where the edges of the strips met came at the same place in the width of the belt, the result was a plane of weakness which, if it came just at the place where the belt bent for troughing, would lead to the destruction of the belt. As a result of knowledge on this point, belts as now made by careful manufacturers have the longitudinal seams or joints at the middle of the belt or near its edges where they will not be affected by the bending of the belt over troughing idlers of the types in general use.

The connection between the steep angle of troughing and the splitting of belts was quite apparent and since the latter was a serious difficulty, it was not long before the angle was changed by Robins and others from 45° to 35°, while later idlers were made with the angle 30°, 25°, 20° and even less. These changes helped belts in still another way, i.e., by reducing the internal stresses in the belt due to the shift of the load in passing over the idlers. In Fig. 38, *A* represents the edge of the load on a belt troughed at 45°. Midway between idlers, the edge of the belt drops and the load slips to a line *B*; this readjustment of the load extends in a smaller degree to the material back from the edge of the belt; some of it is on the surface where it can be seen, but some of it is within the cross-section where it is not noticed. The action of the inclined pulleys in pushing back the load is

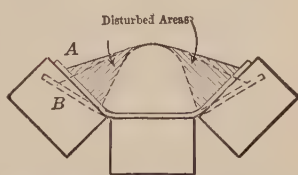


Fig. 38.—Disturbance of Load Cross-section on 45° Troughing Idlers.

exerted through the belt, at the bend it produces a tensile stress on the filler (crosswise) threads in the fabric, and naturally, the less the angle of troughing the less the disturbance of the load and the less the stress in the crosswise threads in the belt.

On the loss of power due to change of cross-section, see page 72.

Belts Running Crooked.—It was a common experience with old-time belt conveyors that the belt would not conform well to the contour of the 45° and 35° troughing idlers, but would tend to lift off the horizontal pulley when lightly loaded or empty. In doing this the belt was apt to be cut by the edges of loading chutes and skirt-boards and by losing the guiding effect of the horizontal pulley, the belt would run crooked. As has been stated, it was necessary to use side-guide idlers to keep the belts centered on these idlers, and belts were often damaged by the constant pressure against their edges. A later chapter explains why the tendency to run crooked decreases as the angle of troughing is made less. This connection was not generally recognized in practice but it is a fact that when idlers with angles of 30° and less came into use, the troubles with belts running off center became less, there was less need for side-guide idlers and it was found that belts of uniform cross section (straight-ply belts) could be used as well as those in which greater flexibility and better contact with the horizontal pulley were attained by omitting plies of fabric near the middle of the belt (stepped-ply belts).

Multiple-pulley Idlers.—Quite early in the belt-conveyor business there were suggestions that if idlers were made with more than three pulleys set to maintain the same depth of troughing as with three-pulley idlers, the bending action on the belt would be easier and the conveyor would suffer no loss of carrying capacity. By the use of a greater number of pulleys, the belt would be bent through a smaller angle along each line of longitudinal flexure, the internal stress on the filler threads of the belt fabric would be diminished, there would be less risk of pinching the belt between pulleys and the tendency to crack or split the belt would be less. The complication of parts and the increased difficulty of lubricating the pulleys were obvious disadvantages which delayed the introduction of multiple-pulley idlers and they did not come into use until after 1905.

The first United States patent on a multiple-pulley idler was that granted to Lynch (1899) which shows an impractical device consisting of six pulleys mounted on a bent shaft. Plummer in 1903 patented a five-pulley idler like those now in use except that the faces of the pulleys were concaved to match the curve of the belt and avoid all angular bends. Neither of these schemes ever came into practical use.

The four-pulley idler, used to some extent fifteen years ago (Fig. 8), is open to the objection that the center of the belt, which is under the greatest depth of load, is apt to sag between the middle pulleys and be rolled into the gap and split along its whole length. The Robins idler of 1909 (Fig. 39) and the Peck idler of 1913 are alike in using five pulleys with the inclined pulleys set at 15° and 30° from the horizontal. Most of the multiple-pulley idlers in use at the present time are made with this arrangement of five pulleys. They differ

chiefly in the ways of making and assembling the cast-iron brackets that support the pulley shafts.

Expedients to Avoid Cracking of Belts.—The splitting of belts by longitudinal cracks was a serious matter in the early days of the belt conveyor business, and a number of schemes were brought forward to prevent it. Some of these referred to belt construction, others to idler construction. Some have been referred to on previous pages. Of those named below, not one is in commercial use to-day; but they are frequently mentioned, and occasionally some similar devices are newly invented and brought to the notice of those in the business.

Belt Devices.—The Selleck patent of 1902 covers a conveyor belt composed of a central strip and two side sections flexibly united at their adjoining edges by interlocked lacings, the lacings being coils of wire enclosing a flexible thong of leather. In other words, the inventor starts with a split belt to prevent splitting. Ridgway, in his patent of 1902, omitted some of the plies at the bending points of a rubber belt and increased the rubber there, to get greater flexibility. Some belts of this "hinged-edge" design

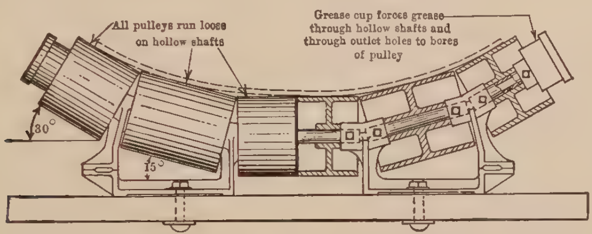


FIG. 39.—Robins' 5-pulley Idler, 1909.

were used in this country and in England; they troughed well and ran straight and in some cases they gave satisfaction. In other cases the belts showed structural weakness along the lines of bend owing to the comparatively few filler (crosswise) threads of fabric there. In other words these belts were half split to begin with; the bending was concentrated along the weak lines and the extra rubber could not hold the belt together. In the three Plummer patents of 1903 the body of a canvas belt is made thick and impregnated and hardened by saturation with drying oils, while the wings are made of fewer plies, and being treated with non-drying oils, remain flexible. The Ridgway patent of 1904 discloses a double belt structure in which the load is carried on a troughed belt which is supported by and receives its trough shape from saddle-shaped cleats mounted on a flat belt running underneath over its own separate pulleys. This scheme was costly and complicated and did not get beyond the stage of advertising and demonstration. Plummer, in 1906, patented a canvas belt in which the upper plies, which take the abrasion of the material carried, are stitched together for their full width, but fastened to the lower plies at their longitudinal central portions only, in such a way as to make the belt flexible for troughing. This belt never came into practical use.

Idler Devices.—The Mann and Neemes idler of 1902 consisted of a spool idler made in five or more sections, the middle section tight on the straight through-shaft, the outer ones loose. It was intended to give the troughing of a deep spool idler without the disadvantage of wear on the under side of the belt (see page 11). It would do this if each loose section of the pulley were sure to turn freely on the shaft, but since lubrication through a long hole with many side outlets is uncertain, it is safe to say that some of the pulleys would not turn freely and the belt would rub and run crooked. Besides that, the construction is apt to be expensive if well made. The Rouse idler of 1907 represents another effort in the same direction. Patents have been issued at various times since 1907 on " helical " idlers in which the supporting pulley is made of a length of steel rod or ribbon wound into a helix of cylindrical shape. In the Thomas idler of 1907, the helix forms a tension spring that bends to conform to the weight of the belt and its load, the ends of the rod being extended on the axis of the helix to enter ball-thrust trunnion bearings carried by a stand at each side of the conveyor. One conveyor equipped with these idlers was set up in Chicago; when a load was put on the moving belt the springs vibrated and set up a wave-motion in the belt which made it inoperative. Vrooman, in 1909, patented an idler of similar form with a number of separate pulleys fitted to a flexible-jointed shaft with a spring-mounted end-thrust bearing at each side. This had the defects of the Thomas idler with some added complications. The same may be said of the Proal patent of 1911, in which it was proposed to use separate pulleys joined together by axial helical springs and carried in end-thrust bearings similar to the above.

The old difficulties with the cracking of belts were lessened, or avoided not so much by radical changes in belt construction, or by novelties in idler construction as by reducing the angle to which the belt is bent along the line of flexure in troughing. This led to the use of three-pulley idlers with the two bends 30°, 20° and as low as 10° and to the use of five-pulley idlers where each of the four bends is 15°. The tendency of the present day is in the same direction, toward wide belts, shallow troughing and simplicity in idler construction.

Efforts to Resist Cutting and Abrasion.—Early experiments by Robins showed that rubber specially compounded resisted the action of a sand blast much better than did other materials, the wear being in the following ratios: Rubber, 1.0; rolled steel, 1.5; cast iron, 3.5; various cotton belts, 5 to 9. These ratios have been quoted by some as showing the comparative worth of these materials for conveying gritty substances, but the comparisons are not valid when, as in most belt conveyors, the action is not simple abrasion, but a combination of cutting and abrasion. It is well known that when a belt with a rubber cover is subjected to severe impact of large sharp pieces, the cover may be cut through to the fabric, and dirt and water may get under the cover and into the fabric, whereupon the belt fails by its cover coming loose or the plies separating while the cover may still retain its original thickness. The threads of cotton in a rubber belt are to a certain extent

waterproofed by the layer of "friction" which covers each ply of duck, but the fibers of the threads are not impregnated with rubber and they still retain capacity for absorbing moisture. When a cut penetrates to the duck, capillary absorption will spread any water which enters there, and when the cotton decays from mildew the plies separate or the cover comes loose in a blister.

Metal Reinforcements.—The steeply inclined belt conveyors known as "tailings stackers," used on gold dredges to carry off the waste, are subject to very severe cutting and abrasion and the action of water. The material runs fom fine gravel to rocks weighing 200 or 300 pounds and may include tree stumps, pieces of lumber and anything else picked up by the dredge buckets. Costly belts used in this service have lasted only a few weeks or months, and since they were often scrapped because of wear in the middle while the edges were still good, efforts have been made to add life to these belts by the use of metal cleats riveted to the working face (St. Clair, 1890; Manning, 1908; Cory & Dandridge, 1909), metal staples (Folsom, 1906); flexible woven wire face (Hohl & Schorr, 1906); imbedded wire coils (Pattee, 1916; Heaton, 1909); leather face (Cook, 1906). None of these has come into practical use; there are objections to all of them. Wherever a hole is made through the fabric for a metal reinforcement, water and sand will enter; if the reinforcing pieces are arranged in parallel rows, the belt may crack by concentration of the bending along those lines as it runs over the troughing idlers. Wire reinforcement rusts out as well as wears out; the bond between the metal and the rubber or the fabric is generally imperfect and as the parts separate under the stresses of troughing and bending over pulleys, water and grit work their way into the belt.

Other Protective Devices.—Another protective device is shown in the Vaughan patent of 1905, where the main conveyor belt is partly covered by a separate narrower belt that acts as a protector to the middle of the wider belt. In the Voorhees patent of 1906, a belt has a rubber cover in which are imbedded cotton fibers set to expose their ends to the impact and abrasion of the material carried. Plummer, in 1906, made a compound belt with a solid woven cotton back for flexibility, with a wear-resisting face of stiff canvas stitched on. In 1916 Bowers patented a rubber belt in which the wearing face contained several edgewise layers of frictioned fabric set on the bias so as to take the wear partly on the ends of the cotton fibers.

Rubber Covers.—In the rubber-belt business as in the rubber-tire business, it has been found that nothing protects the fabric carcass of the belt or the tire so well as a rubber cover or tread. Metal reinforcements and other protective devices add expense and generally fail to protect the body of fabric from abrasion, cutting, and the entrance of water. Rubber covers are depended upon to do that in present-day practice, and in the quality of the covers and their ability to withstand abuse there has been a steady improvement in recent years.

CHAPTER III

BELTS AND BELT MANUFACTURE

RUBBER BELTS

The Duck.—The strength of the belt lies in the duck, a cotton fabric which for conveyor and elevator belts differs from ordinary sail duck or canvas in the fact that the strength of the warp (lengthwise) threads is considerably greater than that of the weft or filler (crosswise) threads. Duck for rubber belt is graded as 20-ounce, 28-ounce, 36-ounce, etc., according to the weight of a piece 36 inches long in the warp by 42 inches wide; the warp threads are usually larger and closer together than the filler threads, and the size of the thread is determined by the number of spun yarns it is made of. Thus, one weave of 28-ounce belt duck may have a warp 6 yarns to the thread, 26 threads per inch and a filler 5 yarns per thread, 17 threads per inch. Another belt maker may use a 28-ounce duck with a warp 6×24 and a filler 5×14, but with a tighter twist in the threads to make up the weight. The strength of the duck depends not only on its weight, but on the degree of twist in the threads, and the cementing action of the rubber depends upon the openness of the weave as well as on the quality of the rubber. All of these factors are important; in the best use of them the belt maker shows his knowledge and skill.

Friction.—The cementing layer of "friction" as it is called in the trade, is a plastic mass of rubber to which is added some sulphur for vulcanization plus certain compounding ingredients which give it that degree of resilience and elasticity combined with resistance to distortion under load which fit the finished product for its particular use and at the same time to the price at which the belt is to be sold.

Manufacture.—The duck is first passed over heated cylinders to remove most of its moisture; then it is run between the lower and middle rolls of a calender press whose three rolls are heated by steam. The plastic mass of rubber is fed between the top and the middle roll; the temperature of the two rolls is so controlled that the compound sticks to the middle roll and since this revolves faster than the lower roll, the rubber is pressed and wiped into the fabric as the duck passes between the two. When both sides of the duck have been treated, the frictioned fabric is cut into strips of the desired width and then assembled into belt. Thus the making of an 18-inch 6-ply belt starts with an inner strip $36\frac{1}{4}$ inches wide folded and rolled to make 2 plies or layers with a butt-joint. Then a strip about $36\frac{5}{8}$ inches wide is folded around with the joint about 2 inches from one edge, then rolled, making 2 more plies; the outer plies are made of a strip 37 inches wide wrapped around the other four with its joint (which is open about $\frac{1}{16}$ inch) located 2 inches from the other edge

of the belt. If the belt is not to receive a rubber cover, the outer joint is closed by a seam strip of soft rubber, see Fig. 40, then the belt is given another pass through a roller press which squeezes all the plies together and rolls the seam strip into and over the outer joint in the duck. Rubber covers less than $\frac{1}{16}$ inch thick are generally calendered on to the outside ply; thicker covers are rolled out in sheet form cut to width and then laid over the assembled plies.

In this condition the belt is soft and "uncured." To vulcanize or cure it, the belt is run between the upper and lower platens of a steam-heated press, and squeezed at the temperature corresponding to 40 pounds steam pressure (280° to 288° Fahrenheit) for 15 or 20 minutes. In the process, a chemical change occurs which makes the rubber less sensitive to changes in temperature, increases its strength and resistance to abrasion, makes it more durable on exposure to the weather and more resistant to chemical reagents. If the temperature of vulcanization exceeds 300°, the cotton is apt to be injuriously affected and the strength of the belt will be reduced.

Most rubber belts are put under stretch in the vulcanizing press before the platens close, and are cured under stretch, as much as 8 per cent of the original built-up length being removed in this way; in some cases the duck may be stretched before it is assembled into belt.

Technique of Rubber Manufacture.—This is a complicated subject; the chemistry of the business is not well understood, although the effects of the various compounding ingredients are well known. There are at least 50 kinds of raw rubber in commercial use, and several dozen compounding ingredients for belts alone. Manufacturers also use several kinds of rubber substitutes made by oxidizing vegetable oils, like linseed oil and cottonseed oil, and there are many grades of reclaimed rubber used in the business, some of which cost more than certain grades of raw rubber. The knowledge of proper compounding and proper vulcanization to fit belts for the many uses to which they are put, comes from long experience, and is one of the most valued assets of the belt manufacturer. Technical works on rubber manufacture give formulas for friction compounds and cover compounds, but they are of no particular use to users of belts. Each belt factory has its own formulas and they are not made public.

The various substances used in compounding rubber are not introduced merely to cheapen the article. Some, like whiting, are inert fillers, but there are others, zinc oxide for instance, which within certain limits toughen the rubber, and increase its tensile strength and its resistance to abrasion. Some ingredients are added to shorten the time of vulcanization, others to soften the rubber to make it more plastic; some of these substances cost more than the raw rubber itself. Compounding is a necessity because raw rubber does not possess in itself all the qualities needed for the friction nor the cover of conveyor and elevator belts. Whether the rubber used is a cheap African rubber costing a few cents a pound, or a high-grade Para or plantation rubber that cost in war time over a dollar a pound, it must be compounded to bring out certain qualities which are needed for the use to which it is put.

As a result of the attention which American manufacturers of rubber belts have given to the compounding of rubber, there has been within the past ten years a reduction in the cost of the compounded rubber and at the same time an improvement in the life and durability of belts, especially in the wearing covers.

Another point aimed at in compounding is to increase the life of the rubber. In its original state, when drawn from the tree, rubber is subject to fermentation and putrefaction like other vegetable juices. These processes are arrested by smoking the liquid as is done in the Brazilian forests or by coagulating it with chemicals as is done with plantation rubber in the East Indies; nevertheless the raw rubber of commerce is to some extent subject to decay by the absorption of oxygen from the air. Rubber bands, which are pure rubber with no addition except enough sulphur for vulcanization, show decided deterioration after a year or two, but the same rubber properly compounded to resist aging, would show proportionately less deterioration, although its original elasticity would be much less than that of the pure rubber.

For the effect of light, heat and age on rubber, see page 44.

Various Rubber Belts.—The duck in elevator belts is generally 32-ounce. Conveyor belts are generally made of 28-, 30-, or 32-ounce duck; special belts for hard work may be built of 36- or 42-ounce duck. For carrying light substances which cause no wear on the belt surface a "friction-surface" belt (Fig. 40) may be used, but in general service, including grain conveying, it is customary to use a belt with a rubber cover. Such a cover in its lightest and cheapest form (Fig. 41), is a sheath of compounded rubber about $\frac{1}{40}$ or $\frac{1}{32}$ inch thick all around the belt; it resists abrasion to some degree, keeps out moisture and helps to prolong the life of the friction rubber. When the material is heavy and the abrasion and cutting severe, the $\frac{1}{32}$-inch cover is not enough; a heavier cover (Fig. 42) is needed to make a belt of balanced construction, that is, one that will get the benefit of the cover until the friction rubber dries out and the plies tend to separate. On the other hand, it is wasteful to use a cover too thick for the service; if the belt in its carcass of fabric should wear out much sooner than its cover, discarding the belt would throw good rubber into the scrap.

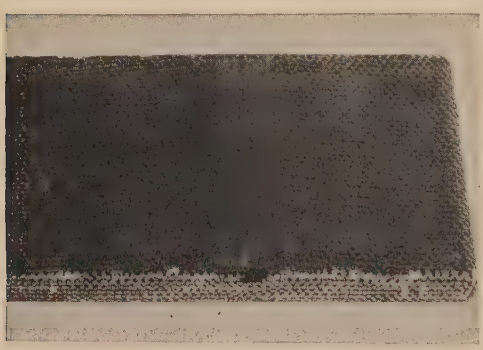


FIG. 40.—6-ply Grain Elevator Belt with Friction Surface, Showing Seam Strip.

Straight-ply Belts.—Most rubber conveyor belts and all transmission and elevator belts are straight-ply belts, that is, of uniform thickness in the

body of fabric. The plies are held together by the adhesion between the duck and the layers of friction rubber; the better the rubber compound, the stronger the adhesion and the longer it will retain its hold on the duck. When the rubber gets old, it loses its tenacity, it tears or separates easily from the duck, and then the plies come apart. The strength of this adhesion is one way to judge the quality of a belt. The friction test is shown in Fig. 49.

Stitched Belts.—When belts are exposed to heat, the friction compound tends to dry out more quickly and lose its hold on the fabric. To prevent the plies from separating from this cause, they are sometimes stitched through and through with cotton twine before the cover is put on and before vulcanizing. There are some drawbacks to this, and the practice is not general because the twine must be waxed for use in the sewing machine, and the cover is more apt to loosen from it than from the duck.

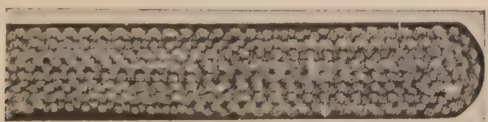


FIG. 41.

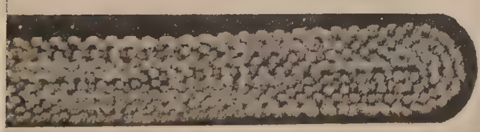


FIG. 42.

FIG. 41.—8-ply Elevator Belt, with $\frac{1}{32}$-inch Rubber Cover on Both Sides.
FIG. 42.—8-ply Elevator Belt with $\frac{1}{8}$-inch Top Cover and $\frac{1}{32}$-inch Cover on Pulley Side.

When the wear from side-guide idlers is severe and the plies are apt to separate along the edges while the body of the belt is still good, rubber belts are sometimes improved by a line of stitching $\frac{1}{2}$ inch from each edge with a second row 1 inch from the first.

Belts to Resist Heat.—Ordinary rubber belts do not hold together well when they carry material at 150° F. or over, especially if the material is fine and holds its heat. If the pieces carried are large, or if the load is a mixture of lumps and fines, the action on the belt is not so bad, because the heat is dissipated more rapidly by the travel of the belt. Rubber belts are made especially for service in hot places or for carrying hot materials; the rubber is not vulcanized to the same degree as in ordinary belts and sometimes a layer of slow-vulcanizing rubber $\frac{1}{32}$ or $\frac{1}{16}$ inch thick is inserted between several of the plies in the assembly. In the finished belt this layer retains its softness and tenacity to a good degree when exposed to heat and prevents the plies from coming apart too soon.

Lamina Belts.—The assembly called by this name in the trade consists in a 6-ply belt, for example, of four layers or plies of duck of the full width of the belt, held together at the edges by narrow strips that form a binding like a book cover, with an outside wrapper of duck which goes over all to form the two outside plies. Such a belt has no longitudinal seams in its body and its edge is two plies thicker than its body (Fig. 43). This makes a stiff edge to resist pressure against side-guide idlers; and, at the same time, the rubber cover is thicker over the surface of the belt than near the edges.

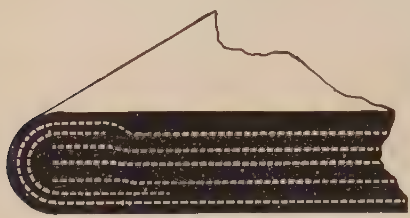


Fig. 43.—" Lamina " Construction of a Rubber Belt.

Stepped-ply Belts.—This style of belt (Fig. 32) has already been mentioned. As now furnished by Robins, the belt is built up in two ways: 1. Rubber cover $\frac{1}{8}$ inch thick in center with 1 ply of fabric less there than at the edges; 2. Rubber cover $\frac{3}{16}$ inch thick in center with 2 plies less there than at the edges. The cover on the carrying surface is about $\frac{1}{16}$ inch thick near the edges in each case.

Stepped-ply belts are made by several manufacturers with various assemblies of plies and covers.

Edges of Rubber Belts.—Wear on the edges of conveyor belts is a common cause of their failure. It may come from the use of side-guide idlers to correct bad alignment of the conveyor or to counteract faults in the design of the troughing idlers. The belt may rub against the delivery chute at the end pulley, or in the tripper, or it may interfere with idler brackets on the return run or with the supporting framework of the conveyor. In rubber belts destruction of the edge exposes the cotton duck to the action of moisture and dirt; the fibers mildew, the friction rubber gives way and the plies separate.

There are several ways of making the edge of a rubber belt. The Metzler edge (patented 1910), used by two manufacturers, protects the outer ply by a thickness of rubber continuous with the top cover (Fig. 44) and has an insertion of rubber under the outer ply to protect the assembled inner plies, so that the latter may have some protection if edge wear should progress beyond the outer ply of the belt. Another method of construction is to bring the top cover around the edge and join it to the rubber cover on the pulley side of the belt (Figs. 42 and 45). On tailings stackers (see page 19) the conveyor alignment is apt to be bad and the belt may have to be held in place by side-guide idlers. The wear on the edge of the belt combined with the wet and grit may make it advisable to protect the edge by an extra thickness of rubber there (Fig. 46). The 5-ply belt shown in Fig. 47 has a very thick edge to resist side wear and a rubber cover $\frac{1}{8}$ inch thick, both of which are held to the body of fabric by a layer of " tie-gum," or cement, in which is imbedded 1 ply of lightweight coarse-mesh fabric known as " cider-press cloth." In making this belt the assembled plies are covered with a layer of strong adhesive rubber,

better than that used in the friction; the open-mesh cloth is laid on top and brought around under each edge for a few inches; then the cover is laid on. When the belt is put under pressure and vulcanized, the layer of tie-gum, or cementing rubber, with the imbedded fabric comes into close and thorough contact with the rubber of the cover and the friction of the duck and holds the two together with a bond stronger than that which ordinarily exists between the cover and the body of the belt. Along the edge the imbedded fabric gives

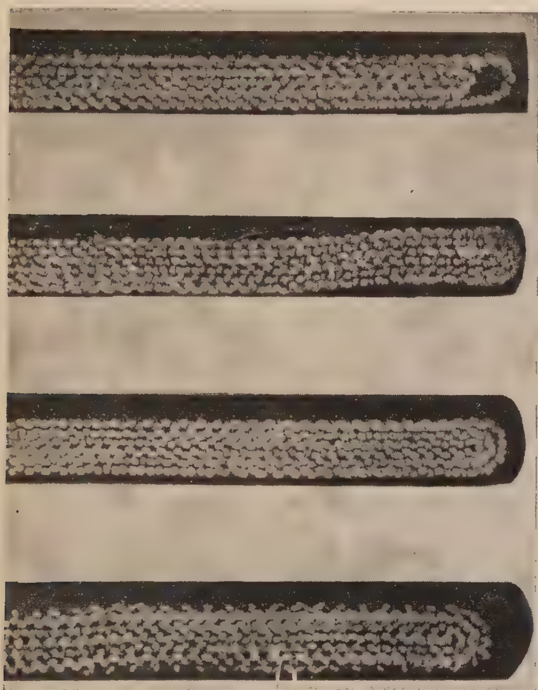


Fig. 44.—(Goodyear Tire and Rubber Co.)

Fig. 45.—(B. F. Goodrich Rubber Co.)

Fig. 46.—(B. F. Goodrich Rubber Co.)

Fig. 47.—(B. F. Goodrich Rubber Co.)

Fig. 44.—5-ply Conveyor Belt with $\frac{1}{8}$-inch Top Cover and Metzler Edge.
Fig. 45.—6-ply Conveyor Belt with $\frac{1}{8}$-inch Top Cover and Rounded Edge.
Fig. 46.—6-ply Dredge Belt with $\frac{5}{32}$-inch Top Cover and $\frac{5}{32}$-inch Rubber on Edge. Cover on Pulley Side $\frac{1}{16}$-inch.
Fig. 47.—5-ply Conveyor Belt Extra Heavy Duck, with $\frac{1}{8}$-inch Top Cover and $\frac{1}{4}$-inch Edge Cemented to Fabric by a Layer of Open-mesh Cloth Imbedded in "Tie-gum." Cover on Pulley Side $\frac{1}{16}$-inch.

some protection to the body of the belt, but its greater use is to form a strong bond between the outer ply and the rubber along the edge, and to prevent the latter from being pulled off in a strip should the belt encounter an obstacle in its travel or meet with an accident in handling or erection. (See also "Reinforced Covers," page 29.)

Rubber Covers.—The purpose of a wearing cover on a rubber belt is to protect the body of fabric, *to an economical degree*, from abrasion and cutting, and from the entrance of moisture. In giving this protection "to an

economical degree," the cover must not be too thin nor too thick. If too thin, the plies may be cut so badly that the belt is thrown away while the friction is still good and before the plies have begun to separate; if too thick, the belt may reach the scrap pile while there is still good rubber in the cover.

If the belt has a good friction and enough plies to transmit the pull easily and safely, and if the operating conditions are good, it pays to use a cover that will develop the full life of the friction rubber; but if the operating conditions are bad, if the belt is likely to be ruined by neglect or accident, if it is too light for the tensile stress, or if the quality of the friction is not good, then it is wasting money to put a good cover on the belt. The proper balance between the quality and thickness of the body of fabric and the quality and thickness of the rubber cover differs according to the material handled, the care in loading and discharging, the hours of service, the length of the conveyor, the importance of the installation and the degree of oversight and maintenance. These conditions are never exactly the same in any two conveyors, and hence there is no general rule for the thickness of rubber covers.

Thickness of Rubber Covers.—In average practice, the aim is to make the cover so thick that the cut made by the impact or drag of a sharp-cornered piece at the loading point of the conveyor will not penetrate through the cover into the fabric of the belt. A cut might injure the tensile strength of the belt, but the greater danger is that water or dirt may get through into the cotton and cause separation of the plies or blistering of the cover. When the cover starts to blister, the trouble is apt to be made worse by the kneading action of the idler pulleys. A blistered cover may catch at a chute or in a tripper and do serious damage.

For run-of-mine coal and heavy ore, covers are generally $\frac{3}{16}$ inch thick.

For crushed coal and ores, covers are generally $\frac{1}{8}$ inch thick.

For fine materials not extremely abrasive, covers are generally $\frac{1}{16}$ inch thick.

Resistance of Rubber Covers to Combination of Impact and Abrasion.—The protection which a good rubber cover gives to a belt can be seen in Fig. 48, which shows the appearance of six samples of belting after a test of one hundred hours made by the author. Two pieces of 6-ply stitched canvas belt, two pieces of 6-ply high-grade rubber belt with $\frac{1}{8}$-inch cover and two pieces of medium-grade belt 6-ply with $\frac{1}{8}$-inch cover were tumbled together for one hundred hours in a cleaning mill with charges of hard iron castings weighing about 750 pounds per charge. Weights were taken at the time new charges of sandy castings were put into the cleaning mill.

The pieces were all 6 inches square at the start and were cut so that one edge in each specimen was the edge of a conveyor belt. The high-grade belt distinguished by the hole drilled in the test specimens showed least loss of size and weight, and the canvas belt lost most. The pulley side of the rubber belts showed more wear than the other side, but the friction rubber offered considerable resistance to the combined action of the sandy grit

and the blows which it received in the cleaning mill. Table 1 gives the record of weights and losses during the one hundred hours.

Quality of Rubber Covers.—There is a direct relation between the tensile strength of rubber and its resistance to abrasion; hence it is possible to secure resisting quality in a rubber cover by requiring it to show a certain tensile strength. The strength of compounded rubber for covers may vary in different belts from a few hundred pounds to over 2000 pounds per square inch; if specifications are written to require a certain tensile strength, the test pieces and the manner of testing should be according to methods which are agreed upon as correct, such as those described in Bulletin 38 of the

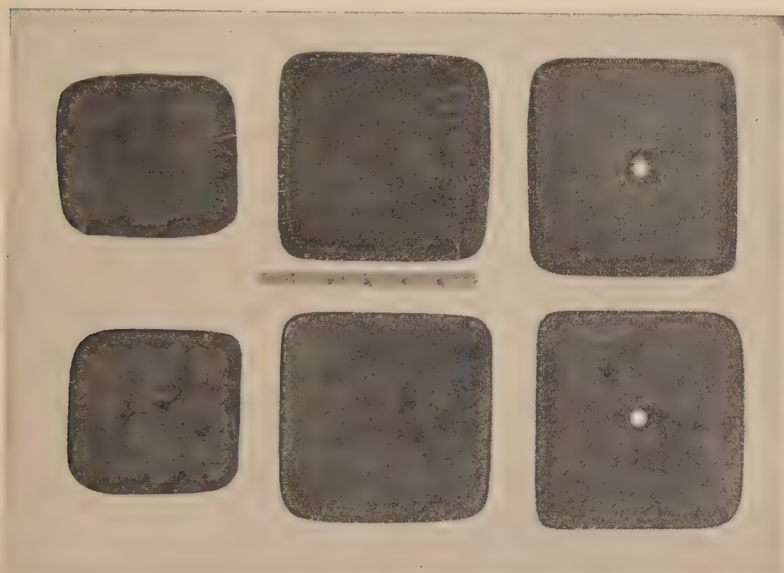


Fig. 48.—Samples of Belt after 100 Hours of Impact and Abrasion.

Bureau of Standards, Washington, D. C. Unless the tests are conducted in such a manner, the results may be open to controversy.

Table 2 (Bureau of Standards) gives the results of tests of four specimens of compounded rubber.

Some specifications for belts carrying run-of-mine coal have required the cover to show 650 pounds per square inch breaking strength, and a strip of it 1 inch wide with marks 2 inches apart is required to stretch to $6\frac{1}{2}$ inches between marks without breaking and without showing more than $\frac{3}{16}$ inch permanent stretch when measured immediately after the test. Another specification for a belt with $\frac{3}{16}$-inch cover, handling heavy ore required a piece to stretch from 2 inches between marks to 11 inches before breaking. The thin covers $\frac{1}{40}$ or $\frac{1}{32}$ inch thick, used on grain belts, should be able to pass the following test. "The outer ply shall be stripped from a piece of belt

TABLE 1.—COMPARATIVE RESISTANCE OF BELTS TO BLOWS AND ABRASION

(Specimens 6 inches square. See Fig. 48)

Hours of Test	Stitched Canvas Belt		Medium-Grade Rubber Belt		High-Grade Rubber Belt	
	Specimen 1, Weight Grams	Specimen 2, Weight Grams	Specimen 1, Weight Grams	Specimen 2, Weight Grams	Specimen 1, Weight Grams	Specimen 2, Weight Grams
At start	203.5	204	305	304.7	305.8	314
7	201	202	301	300	301	310
14	192	191	295	295	296	306
21	185	186	289	290	292	301
28	170	171	280	281	285	295
35	155	155	269	272	279	286
42	140	140	263	266	271	281
49	133	131	257	258	263	270
56	127	124	254	255	257	267
63	115	115	247	247	251	259
71	108	107	244	245	248	257
79	102	101	240	240	245	254
87	93	93	235	236	241	247
100	90	89	234	235	240	246
Total loss in grams.......	113.5	115	71	69.3	65.8	68
Average loss in per cent.....	56.0		23.0		21.5	

TABLE 2.—TESTS OF COMPOUNDED RUBBER

(Bureau of Standards)

Specimen Number........	1			2			3			4		
Pulling speed, inches per minute................	5	25	45	5	25	45	5	25	45	5	25	45
Breaking strength, pounds per square inch.........	2495	2690	2720	1900	1940	1970	375	430	465	340	390	430
Ultimate elongation, per cent..................	605	635	635	465	500	490	340	360	375	105	115	120

and bent over flat on itself with the rubber cover outside, and a second bend made at right angles to the first bend. No cracks shall appear anywhere on the surface when the bent and folded ply is pressed as hard as possible between the thumb and finger." The following is from a specification for belts with $\frac{1}{32}$-inch covers for elevating and conveying sugar and char in a sugar refinery. "A piece of the cover $\frac{1}{4}$ inch wide shall stretch without breaking from 2 inches between marks to 3 inches between marks; upon

immediate release, the piece will rest five minutes and must then not show more than $\frac{1}{8}$-inch elongation in the 2 inches."

From an inspection of Figs. 52 and 53, page 45, it is apparent that the age of the rubber specimen has a great effect upon its tensile strength and elongation. All rubber compounds do not deteriorate alike, as may be seen by comparing *G7* with *G4*. *G4*, when fresh, showed higher than *G7* both in tensile strength and elongation, but was lower in both respects after six months. This point is not covered by any specifications in commercial use. So far as belt manufacturers are concerned, one maker may rate a certain grade of cover so far under its maximum strength that it would pass the tensile test at any time during the normal or expected life of the belt. Another maker might rate the same cover much higher with a fair certainty that it would pass the test in the time between placing the order and acceptance and payment.

It is therefore not an easy matter to get a good cover merely by writing a specification for it. If reliance is placed wholly on the specification, the requirements must be written with skill and wisdom, and the tests made and followed up with exactness and persistence. It is, however, quite practicable to put the burden of responsibility on the manufacturer; there are a number of makers of rubber belts who have long experience in the business and whose skill and knowledge, supplemented by continual test and research, place them in a position to know what a good cover is, and how to make it.

Reinforced Covers.—When the conditions of loading are such that belts naturally wear through the cover near the middle while the edges of the cover are still good, it may pay to use a stepped-ply belt, or a straight-ply belt in which the cover is thickened toward the middle. The original belt of this latter style (see Fig. 31), did not run well on 45° and 35° troughing idlers and was superseded by the stepped-ply belt. In recent years it has been revived by several manufacturers; it can be used to advantage where the abrasion is severe in the middle of the belt and where the depth of troughing compared with the width of the belt is not too great. The extra thickness is made by applying a strip of rubber to the top of the regular cover before pressing and vulcanizing. One maker's standard practice for such reinforced covers is to make the strip $\frac{1}{16}$ inch thick and half as wide as the belt in sizes up to 32 inches. For wider belts, the extra $\frac{1}{16}$-inch thickness runs to within 3 inches of each edge.

Covers with Reinforcing Fabric.—Important belts with very thick covers may have imbedded in the rubber of the cover a "floating ply" of open-mesh fabric or of duck to resist the tendency of sharp heavy lumps to gouge pieces out of the cover. The following paragraphs from a specification issued by a large mining company refer to 48-inch belts in long conveyors handling hard, heavy ore in pieces averaging 5 inches, but ranging up to 15 inches.

"**Rubber Covers.**—All belts to have a covering of high-grade rubber on the carrying side not less than $\frac{1}{4}$ inch thick. Throughout the center part of this cover on the carrying side is to be inserted 1 ply of duck, known as a 'floating ply,' the total thickness of cover and ply to be $\frac{1}{4}$ inch. The cover

on the back of the belt shall be $\frac{1}{16}$ inch thick. The edges of the belt to be rounded and the rubber covering not less than $\frac{1}{16}$ inch thick. The cover on both sides to withstand the action of the dry climate without hardening so that its elasticity and toughness shall not be impaired."

"**Cohesion of Cover.**[1]—There is to be placed between the cover and the first ply of duck, 1 ply of open-woven fabric known as 'cider-press cloth,' this cloth being carried around the edges and imbedded in frictioned rubber, the total thickness of the top cover with floating ply and cider-press cloth being $\frac{1}{4}$ inch. The adhesion between the cover and the cider-press cloth shall be such that in case of accidental contact with projecting machinery, chutes, etc., the cover will break before it tears loose from the fabric. The adhesion between the cider-press cloth and the duck shall show the same friction strength as between the plies, i.e., 20 pounds."

Tearing and Gouging of Rubber Covers.—A strip of good cover stock $\frac{1}{8} \times \frac{1}{2}$ inch cannot be stretched to the breaking point by the strength of a man's hands; but by a tearing action starting at a nick in one edge of the strip, it can be pulled apart with the fingers. Covers are apt to be torn in this way by sharp pieces jamming under skirt-boards that are set wrong, by the chute filling up at the discharge end or in a tripper, stopping the conveyor while loaded so that some material dribbles over a pulley and catches between the chute and the belt, by steel chutes or tools falling down on the belt, or by carelessness in handling the belt in the roll or in pulling it into place over the idlers. Accidents like these will tear any kind of rubber cover, good or poor. If large pieces of the cover rip loose from the top ply with a clean separation, the belt may be defective, but it should not be condemned because the cover does not resist gouging or tearing.

There is no direct ratio between the tensile strength of rubber and its resistance to a tear or shear, although in general, a high tensile cover stock is less likely to tear than one of inferior strength in tension. There is apparently a contradiction where the specification quoted above calls for 20 pounds friction pull per inch (see also page 35) between cover and top ply and at the same time says that the cover must break before it tears loose. It is, however, a valid requirement, because while the tensile value of a strip of cover 1 inch wide is much greater than 20 pounds, nevertheless, in any good belt, the cover will tear before it will pull loose. It is better that it should; a tear can be cemented and patched, but a loose cover can never be fastened on again to stay.

Specifications for Rubber Belts.—Rubber belts came into use for handling grain about 1870; for years after that, they were generally sold on the representations of their makers as to quality. The technique of manufacture was not well developed in the early years, but some of the belts were of remarkably good quality. The late Samuel W. Neall, who had charge of the Washington Avenue Grain Elevator in Philadelphia, for many years, related the following: When the two galleries on the pier were erected in

[1] Fig. 47 illustrates the "tie-gum" construction described in this paragraph. See also Fig. 239.

1873, two 36-inch belts about 800 feet long were installed. In 1900 the two galleries were torn down and replaced by a single one twice as long; a new 36-inch belt was put in and, parallel to it, a 1600-foot belt made by joining the two old ones. The new belt lasted seven years, the old belts were still there and in use when the elevator was dismantled in 1916, forty-three years after they were originally installed. When the Pennsylvania Railroad Company's Girard Point Elevator was torn down in 1916, the original 36-inch 4-ply rubber belts put in in 1882, 34 years before, were still in regular use. They were 700 feet long, and ran flat with portable concentrators at loading points only.

As the business grew, the number of belt manufacturers increased, and competition for business brought on the market many belts of poor quality. Some makers probably did not know how a good belt should be built. In the grain business, this led to the use of detailed specifications for rubber belts. A typical set of specifications is that issued by the John S. Metcalf Co. of Chicago. The following is the 1913 edition:

METCALF'S SPECIFICATIONS FOR GRAIN BELTS

1. The belts shall be first class in quality, of standard manufacture, fully up to the grade specified and of a light gray color. All duck used in the construction of the belts shall be of the best quality. In all belts a piece of duck 36×42 inches shall weigh not less than 32 ounces.

2. The tensile strength of a piece of duck 1 inch wide shall not be less than 350 pounds in the direction of the warp. This shall be the average strength of the entire width of duck tested in pieces 4 inches wide or wider, with the jaws of the testing machine not less than 1 inch apart.

3. The several plies shall be thoroughly cemented together with first-class quality of Para rubber compound, making such an adhesion between the different plies of duck that it will stand the following test, viz.:

4. The test shall be made by cutting a strip 1 inch wide, running longitudinally of the belt. The various plies shall be separated at one end and the strip suspended by attaching by its upper end, and a weight of 10 pounds shall be attached to the lower end of the strip to keep it in a vertical position. Then the other plies shall be pulled off, one at a time, by attaching a 13-pound weight to the upper end of the outer ply. A coil spring shall be placed between the upper end of the ply tested and the weight. No test shall be made between the last two plies on the strip.

5. Under the above conditions the plies must not separate faster than at the rate of 2 inches in one minute. The cementing compound when pulled apart shall show there is ample rubber in its composition to insure its lasting quality, and it shall also show a long, clinging, fibrous adhesion to the duck.

6. **All duck used in the manufacture of the belts shall be of the widths, also the joints located only, as shown on blue printed sheet attached. (Fig. 50.)**

7. The outside covering shall be a coat of rubber not less than $\frac{1}{40}$ of an inch thick, evenly put on, thoroughly vulcanized and finished.

8. Shop splices across the belts shall not come closer than 10 feet to the field splice in any conveyor belt.

9. The belts shall be straight, well stretched and pressed. A sample of the belt not less than 24 inches long and 22 inches wide must be submitted to the Engineers, John S. Metcalf Co., for their approval before the belts are manufactured; also a sample of duck 36×42 inches. These samples shall receive the written approval of the Engineers before the belts are manufactured. The samples approved in writing by the Engineers will be held by them to compare with the belts when delivered. If they are not equal to the samples they will not be accepted.

10. The Engineers may cut into the belts at any point and make such tests as they may consider necessary in order to determine whether the quality of the belts is uniform and up to the grade specified.

11. The testing of the belts will be done within three months after the time the belts arrive on the ground and belts must conform to these specifications, and to the samples submitted and approved, at any time they are tested within said three months.

12. The belts furnished shall be equal to these specifications and also equal to the samples submitted and approved. If the belts as furnished do not come up to the specifications, or samples submitted and approved, the Company shall have the privilege of running them until new belts are supplied and can be installed without inconveniencing said Company; and the manufacturer shall at his own expense, take them out at such time as may be ordered by the Company and replace them with belts that do conform to these specifications, and are equal to the samples submitted and approved.

13. The manufacturer shall guarantee that the belts will not blister, nor separate in the plies or at the seams, within one year from the date of installation; and shall guarantee to furnish and install new belts, complying with these specifications, if such defects occur; and shall take out the defective belts at his own expense. The manufacturer shall not be allowed anything for the service obtained from belts which are replaced by other belts.

14. All of the elevator belts shall be accurately punched for bucket bolts in rows across the belt; the rows being at right angles to the length of the belt.

15. The Engineers, or the Company, shall supply the manufacturer with a drawing showing the size and spacing of the bolt holes, or shall furnish a template for this purpose.

16. In the above specifications Engineers mean John S. Metcalf Co.; Company means the Company owning the elevator in which the belts are to be installed.

Comment on Metcalf's Specifications.—Since these specifications and similar ones based on them are frequently referred to in buying rubber

belts, the following comment on the numbered paragraphs may be of interest.

¶ 1. " First-class in quality " should be qualified by the words " for the purpose intended." Belts for coal and ore are generally of higher " class " than grain belts. " Light gray color " refers to the appearance of the finished belt with the " bloom " still on it, not to the color of the rubber in the belt as some persons have thought.

¶ 2. It is not a simple matter to test the strength of belt duck. If a test is required, the method of making it should be agreed upon. In recent years most belt manufacturers test their duck before cutting it up for belts, but their methods of testing are not all alike and their numerical results are seldom comparable. The following statement is by the Goodyear Tire & Rubber Co., Akron, Ohio:

METHOD OF TESTING FABRIC AS EMPLOYED BY THE GOODYEAR TIRE & RUBBER CO.

There are a large number of methods in use for testing the tensile strength of fabric and finished belts, the most of which have received some recognition. The same piece of duck may show a tensile strength of 300 pounds or 700 pounds or any intermediate value, depending on the method chosen. When the same duck is built into a belt and the belt tested, an entirely different tensile strength value may be obtained. This is due to the fact that the same methods of testing are not adapted to testing both raw duck and finished belts, rather than to any appreciable change in the strength of the duck itself during the building operation.

Therefore, whenever referring to any such tensile strength figures, it is absolutely imperative that the exact details of the testing methods be specified.

None of the methods which have been devised up to the present time definitely determine what we refer to as the " true strength " of duck, although much research has been done on this problem. The most practical methods are known either as " Strip " or " Grab " methods.

Strip tests are made on specimens raveled down to a definite width, usually 1 inch wide or else to a definite number of threads. All four jaws of the testing machine must be wider than the sample in order to thoroughly grip all the threads to be tested. This method gives results which are lower than the " true strength " because the outside threads pull themselves free from the rest of the fabric.

The Bureau of Standards advocates a Strip method and it is probable that it is slightly more accurate than any of the Grab methods, but their method is not generally adopted because of the great expense in preparing the samples.

Test specimens for the Grab method are cut wider than the jaws and are not raveled to any definite width. One of the four jaws of the Test machine is usually 1 inch wide, and the other three jaws are usually 2 inches wide,

thus grabbing firmly a section of duck exactly 1 inch wide. The results by this method are higher than the "true strength" because in reality more than 1 inch wide of fabric is under tension. However, the results by this method are fully as comparative as by the Strip method.

Goodyear tests every roll of belt fabric, and those rolls which fail to meet a definite standard are rejected. Inasmuch as at least six specimens are necessary from each roll (three from the warp and three from the filler), it is found that the Strip method is too expensive for the slight advantage gained by this method over the Grab method. It has, therefore, been necessary for us to use a Grab method and the following details are standard with us:

The specimens are cut $2\frac{3}{4} \times 6$ inches and placed in a Scott Fabric Testing Machine, equipped with a pair of 2-inch wide jaws on one end and on the other end, a 1-inch jaw bearing against a 2-inch jaw. The two pairs of jaws are 3 inches apart at the start of the test and are separated at the rate of 20 inches per minute. The machine indicates the maximum tensile at the time the fabric breaks. This figure is then corrected for moisture.

Cotton before being frictioned absorbs moisture from the air very readily. The Bureau of Standards has found that the tensile strength of cotton fabric increases 7 per cent with every increase of 1 per cent in the moisture contents. The strength decreases again with the drying out of the fabric in the same proportion; therefore, we determine the moisture content on every sample of duck and the tensile strength is corrected to the standard basis of 6 per cent moisture by the above method.

¶ 3. "First-class quality"—the 13-pound friction test called for in the next paragraph really specifies the quality. Friction that tests 18 or 20 pounds is of higher quality. "Para" rubber from Brazil was once the standard of quality; now over 75 per cent of the world's rubber comes from plantations in Ceylon, India, and Sumatra, and as to price and quality it averages higher than Para rubber from Brazil. As to specifying Para rubber, the following is to the point.

"Many consumers do not appear yet to understand that there are many other qualities of rubber besides fine Para, and that for many purposes some of these rubbers may be equally satisfactory to, and in others more satisfactory than, pure Para. For instance, where great strength and resilience are required pure Para cannot be bettered, but even in this regard there are certain rubbers, other than Para, which are for many practical purposes sufficiently strong and resilient, yet which are of decidedly lower price. In other instances fine Para may actually be less desirable than a lower class rubber; thus in certain cases it is desirable to have a rubber which contains rather more resin than the fine Para. A consumer, therefore, who, without reference to his actual requirements, strives to pin down the manufacturer to fine Para, will as likely as not receive an inferior article for his purpose compared with that which he might have had if he had given the manufacturer a free hand in this regard. It may be inferior for a variety of

reasons. In the first place Para may be unsuitable for the article, or the high price of Para may conduce to an inclusion of an excessive amount of mineral matter, not to speak of remade rubber or rubber substitutes, in the article. Again, the term 'best rubber' may mean anything or nothing. A consumer may thereby mean Para or merely rubber unmixed with any other hydrocarbon. The term 'properly vulcanized,' which is used in many specifications, is likewise, in my opinion, an objectionable one, as it is capable of giving rise to interminable controversies, and is in itself a suggestion that the manufacturer is likely to shirk the most important part of his business."[1]

¶ 4. A simple way to make the friction test is shown in Fig. 49: lines 1 inch apart are marked on the strip; the plies are carefully separated for an inch or two and a 10-pound weight is hung to the lower end of the strip to keep it vertical. The pulling weight which is to measure the friction pull, whether it be 10, 13, or 18 pounds or more, is attached to one ply and the time rate at which it pulls loose is noted. The adhesion between the last 2 plies is not measured because in spite of the 10-pound weight, the plies will bend at the point of separation and the pull will not be at 180° to the plane of the separating ply. The long spring is to steady the pull and prevent jerks.

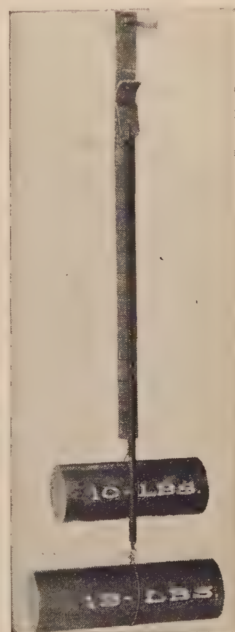


Fig. 49.—Test of Quality of Friction Rubber by Determining the Rate of Separation of Plies under a Given Weight.

The time rate of separation is an important factor in these tests. It never varies directly as the pull. Up to a certain pull, there may be no separation at all, after which the rate increases gradually, then more rapidly, and finally a very small increase in the pull causes a great change in the rate of separation.

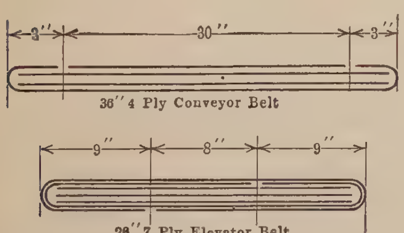


Fig. 50.—Assembly of Duck to make Certain Belts according to Metcalf's Specification.

Bulletin 38 Bureau of Standards 1921 edition describes a power-driven "friction" testing machine in which the pull required to separate the plies and the time-rate of separation are automatically recorded on a chart.

¶ 6. This refers to the construction shown in Fig. 50, which for the particular job illustrates the assembly of 36-inch 4-ply conveyor belts and 26-inch 7-ply elevator leg belts. The point aimed at in the conveyor belt is to avoid longitudinal seams at places where the belt bends on the trough-

[1] From "Rubber," by Philip Schidrowitz, Ph.D., F. C. S., London 1911, pp. 292–3.

ing idlers; there are no joints in the inside plies and those in the outside ply are about 3 inches from the edge where no bending ever occurs. In the elevator leg belt the seams are kept away from the edges and also from the middle where the belt bends on the crown of the pulleys. Belt-makers have been known to make up the width of the inside plies of belt by assembling narrow strips which would otherwise be waste from the cutting operation. Such belts will not trough without cracking. At one of the Western smelters a heavy 42-inch belt failed by longitudinal cracking after a few months' use. Examination showed that all the inner plies were of the full width of the belt, but the wrapper which formed the two outside plies was butted at a place which happened to come over the gap between two pulleys of the troughing idlers. This gap was $1\frac{1}{8}$ inches wide. The point of weakness in the outside ply localized the bending there; the belt sagged into the gap, was jammed between the pulleys and was ruined.

¶ 7. A rubber cover $\frac{1}{40}$ or $\frac{1}{32}$ inch thick is the standard minimum for rubber-covered belts.

¶ 8. "Shop splices" refers to piecing the duck end to end in making up the belt. Piecing is done to use the duck economically and because the strips as received from the duck mill are never over 550 feet long in the roll. A butt splice in a ply close to a field joint might prevent making a good step-splice or it might pull loose under load. Belt specifications have called for belts over 550 feet long without end splices in the duck, but with looms built as they are, it is not possible to get the fabric in such lengths.

¶¶ 12 and 13. These paragraphs may have been necessary in dealing with inexperienced or unscrupulous manufacturers, but reputable belt-makers object to some of the clauses. The explanation offered on behalf of the specification is that the engineer is always the arbiter and that cases calling for rigid enforcement of these threats and penalities are very rare in recent years; and that in case of belt failure the engineer would decide whether it came from poor quality or an original defect, or as a result of bad alignment, excessive tension, defective splicing or some other negligence or error on the part of the user.

Strength of friction is in itself no sure index of the quality of the belt; if it were, the plies of a fabric belt might be glued together and show a very high test, but such a belt would fail by cracking of the glue. In a similar way a belt can be made with a rubber friction compound that will show a high test when fresh but which will not keep its strength six months. For instance, a low-grade friction "doped" with rosin or shellac may show 18 pounds when fresh, but less than 10 pounds when ten months old. Then, too, the adhesion of the friction depends upon care in manufacture as well as on the inherent strength of the rubber compound. The openness of the weave of duck, the degree of twist in the threads, the percentage of moisture in the duck when the rubber is pressed into it, the freshness of the frictioned surfaces when pressed together, the freedom of these surfaces from dust and the pulverized soapstone so freely used in rubber factories—all these factors are under control in the best establishments; but still all that the

manufacturer can hope for is that his friction tests for a certain duck and a certain compound will fall within a range of 10 or 12 per cent above or below the desired standard. It would be better if specifications permitted such a range in the friction tests; when a definite pull is insisted on, the only safe way for the experienced and careful belt-maker is to call that figure his minimum and make his belt to average a few pounds higher. There always have been concerns in the belt business with a reputation to make and none to lose; with some of these the definite specification offers a temptation to go after the business with a low price on a belt that may " get by " on a high pull. Another circumstance which has had its effect on low bidding is the knowledge in the trade that belts are usually wanted soon after the order is placed and when once in service they have more than an even chance of staying there in spite of specifications because a replacement often means annoyance and loss to the user beyond the value of the belt.

With the growth of technical knowledge in the belt business there has come greater co-operation between makers and users. Specifications for grain belts have been changed from time to time and the standard of quality has been raised. Some engineers prominent in the business buy rubber belts on the reputation of the makers without detailed specifications; others have written specifications which in essentials are a restatement of the makers' own ratings of their trade-marked belts as to weight and strength of duck, strength of friction and make-up of plies.

The following is a recent specification issued by James Stewart & Co., Ltd., Chicago, for grain belts with 16-pound friction:

SPECIFICATIONS FOR RUBBER BELTS
JAMES STEWART & CO., LTD.

" 1. **Rubber Belts.**—All belts shall be made of several plies of cotton duck cemented together and covered with rubber compound and shall be straight, well stretched and pressed, first-class quality, of standard manufacture and fully up to the grade specified. The elevator leg belts shall be 6-ply; and conveyor belts shall be 4-ply.

" 2. The duck used in the construction of these belts shall be of the best quality of Mount Vernon or equal grade of cotton duck. A piece of duck 36 × 42 inches shall weigh not less than 32 ounces for elevator belts and not less than 30 ounces for conveyor belts. The duck for each belt shall be made in one piece lengthwise without any other splice except one splice to make the belt endless for all belts less than 450 feet long. The tensile strength shall be tested by pulling a strip 2 inches wide cut from finished belt longitudinally by a fair pull in the jaws of an approved testing-machine. The ultimate tensile strength shall be at least 320 pounds per inch wide per ply for belt made of 32-ounce duck and shall be at least 300 pounds per inch per ply for belt made of 30-ounce duck. The joints of the outside ply of duck for the conveyor belts shall be 2 inches from the

edge of the belt and the joints in the elevator belts shall not be nearer than 5 inches to the edge of the belt.

"3. The several plies shall be thoroughly cemented together with first-class-quality rubber compound making such adhesion between the different plies of the duck that they will stand the following test:

"4. A strip 1 inch wide shall be cut longitudinally from the belt and suspended in a vertical position by attaching the upper end to a hook and fastening a 10-pound weight to the lower end. The various plies will then be pulled off one at a time; first, separating the ply for a distance of 2 inches and then attaching a 16-pound weight to the ply, a coil spring scale being placed between the weight and the ply tested. Under the above conditions the separation of friction must not be faster than at a uniform rate of 1 inch in one minute. The cement compound when pulled apart shall show there is ample rubber in the composition to insure lasting quality, and it shall also show a long, fibrous, clinging adhesion to the duck.

"5. The outside covering shall be a coat of rubber not less than $\frac{1}{40}$ of an inch thick, evenly put on, thoroughly vulcanized and finished and shall stand the following test: The outer ply shall be stripped from the belt and bent over flat on itself with the rubber cover outside and a second bend made at right angles to the first bend. The cover shall be so elastic and so tough that no cracks shall appear anywhere on the surface when the outer ply is pressed as hard as possible between the thumb and finger.

"6. Samples of each kind of belt 3 feet long and of full width and a sample of each weight of duck 42×36 inches shall be furnished after order is placed. These samples will be held by the engineer to compare with the belts when delivered and if they are not equal they will be subject to rejection.

"7. The belts furnished shall be equal to or superior to these specifications and the samples submitted.

"8. The manufacturer shall furnish experts and tools and shall splice the conveyor belts with even step splices in place thoroughly cemented and stitched together under the direction of the engineer.

"9. Elevator belts shall be punched by manufacturer for bucket holes in rows across the belt at right angles to the length of the belt as shown on the detailed drawings."

Comment on Stewart's specifications:

¶ 1. "First-class quality" should be qualified by "for the purpose intended." Sixteen-pound friction test represents an advance beyond the old standard 13- or 14-pound test, but some recent belts, notably those for the Pennyslvania Railroad Co.'s 5,000,000-bushel Elevator No. 3 at Baltimore, have been made to test still higher, i.e., 20 pounds, not that the higher grade of friction adds any useful physical quality to the belt, but that it will keep its "life" longer. Belt-makers consider that with proper care a belt with 16-pound friction should average over ten years in grain service before the rubber loses its "life" and the belt fails by separation

of the plies; belts with lower-grade frictions are not expected to last so long, but a belt with 20-pound friction should last twenty years or more, barring accident or abuse.

¶2. The Goodyear Tire and Rubber Co. makes the following statement as to tensile tests of finished belt:

"Many methods for testing finished belts permit the breaking of the sample to occur in the jaws of the testing-machine and are, therefore, not fair to the sample. To obviate this, we have designed a sample 4 inches wide at the ends and constricted to 3 inches wide for a length of 11 inches in the middle with a total length of about 22 inches. A 1-inch radius is used where the constricted part joins the wider part for the jaws. We combine a stretching test with the tensile test by holding on the sample a load equivalent to one-third of the total specified strength of all the duck in the constricted part of the sample for a period of five minutes and then report the stretch in per cent. The load is then increased until the sample breaks, the increase being at the rate of 2 inches per minute.

"A wide discrepancy is to be expected between the belt strength and the duck strength when tested according to the above methods. There are three important reasons for this, namely: (1) Inasmuch as the Grab test gives higher strength test figures than represent the true strength of the fabric, the testing of a belt containing, let us say, 12 inches wide of fabric in the constricted portion will not show 12 times the strength of 1 inch of fabric by the Grab method. (2) In spite of our efforts to build belts with all the plies perfectly co-operating, we have not eliminated this factor from our product. It often happens that some of the plies are under slightly more tension than others so that when the sample is placed in the testing-machine, these plies, having the higher tension, are broken before the other plies are quite up to their maximum tensile strength. (3) The question of speed also enters into this problem. In general, in testing, the higher the speed, the higher the test results on the same material. The most uniform results on duck-testing are obtained at fairly high rates of speed which are not practicable for the testing of large pieces of finished belt because the latter must be tested in very large testing-machines.

"We should also mention that some prefer pulling the duck from finished belts and testing the duck in one of the usual manners. In this way, very poor workmanship in obtaining the co-operation of the various plies would not be discovered."

Another reason for expecting, and allowing for, some variation in tests of strips cut from belts of the same grade, or even from the one belt, is that it is impossible to trim a test strip to an exact width without cutting through one or more warp threads in each ply along the edge of the strip. If the belt is made of duck with 24 warp threads per inch, a 2-inch-wide test strip might have in each ply 50 or 48 or even only 44 effective warp threads to take the load. Between these there is an unavoidable variation of 12 per cent. If the test strip were 3 inches wide, the same differences in the

number of effective threads would mean a variation of only 8 per cent. The wider strip makes a fairer test.

¶ 4. The rate of ply separation depends, of course, on the adhesion of the rubber compound to the duck. A friction that tests 16 pounds will naturally strip more slowly than a 13-pound friction; a 20-pound friction on 42-ounce duck might not strip faster than 8 inches in ten minutes.

¶ 5. "Outside covering $\frac{1}{40}$-inch thick." This is standard (see Fig. 41) for the back cover on many belts for coal and ore and for top and back covers on grain conveyor belts. There are standard elevator leg belts which have a "friction" surface, that is, no extra rubber cover, but merely the thin layer pressed on to the duck in the calendering process. See Fig. 40.

¶ 8. The construction of a step splice is shown in Fig. 60.

Other Specifications.—Some of the requirements mentioned in other specifications for rubber belt are as follows:

1. Belts 20 inches or less in width shall have only one seam in the cover ply.
2. Belts over 20 inches wide may have two seams (in order to avoid the use of duck wider than the 42-inch standard), but the seams must not come in the center of the belt nor near places where the belt bends in troughing.
3. Belts will be rejected for defects in workmanship, such as:
   (*a*) Open seams or plies not meeting, requiring a narrow filler strip to close the gap.
   (*b*) Blisters between plies or under the covers, repaired or not repaired. (These may be due to moisture in the duck at the time of vulcanizing.)
   (*c*) Rubber cover torn off edges of belt (careless handling in manufacture).
   (*d*) Soft edges, i.e., outer plies not backed up by inner plies.
   (*e*) Cover ply broken by overstretching.
4. Tests to be made on finished and delivered belts. The following paragraphs refer to an order for belts to be used in a sugar refinery for conveyors and elevators for raw and refined sugar and for barrel elevators.

"**Friction.**—A 1-inch-wide sample will be cut crosswise from the belt, the cover opened at the seam and each ply tested singly as to its union with the next ply by measuring the rate of separation caused by a weight of 14 pounds hanging freely to the upper end of the ply under test, as the sample hangs against a wall. The rate of separation must not exceed 6 inches in ten minutes for any one ply nor average more than 4 inches in ten minutes for all the plies."

"**Belts Spliced Endless by the Manufacturer.**—The purchaser reserves the right to cut across an endless conveyor belt and remove a strip 1 inch in width for testing purposes. In the event that this test is made by the purchaser and the belt passes the prescribed requirements, the ends will be joined together by fasteners and it will be acceptable. If the belt fails to fulfill the prescribed requirements, it will be rejected, and it is mutually

understood that the contractor will have no claim for damages against the purchaser for damaged belt or belts due to the cut."

Location of Transverse Splices in Duck.—The following paragraph is taken from a recent (1922) specification for 48 inch 8- and 10-ply rubber belts used for long coal conveyors.

"Belts will be furnished in roll lengths of approximately 500 feet to allow the economic use of duck with as few transverse splices as possible. In no case shall there be any transverse splices in either of the outside plies of the belt in a 500-foot roll. When transverse splices are necessary in the inside plies, they shall be made by cutting the ends of the duck at 45 degrees, and no two splices shall be closer together than 15 feet in the run of the belt."

Improvements in Rubber Belt Manufacture.—The engineering and manufacturing staffs of the companies that make rubber belts have effected great improvements in the design and manufacture of the cotton duck, in the compounding of the rubber and in the application of rubber covers. Further improvements may be expected in methods of making the threads and individual fibers of the cotton thoroughly waterproof, and in bonding the rubber to the duck in such a way that the plies of fabric will be held together more firmly. The Pratt patent No. 1349911 of August 17, 1920, describes a process which is intended to accomplish these results. It proposes to saturate the fibers of cotton duck with a sulfur-terpene solution, then after evaporating the solvent, to "friction" the duck with rubber in the usual way. When the assembled plies are vulcanized, the sulfur-terpene compound is to react chemically with the rubber and form a close bond between it and the cotton fibers.

Weights of Rubber Belts.—Most conveyor belts are built of 28-ounce duck. The cover on the pulley side is about $\frac{1}{32}$ inch thick on widths up to 24 inches and $\frac{1}{16}$ inch thick for wider belts. The cover on the carrying side will be the same as that on the pulley side unless it is ordered thicker.

Table 3 (Goodyear Tire and Rubber Co.) gives unit weights of rubber

TABLE 3.—WEIGHT OF FRICTIONED FABRIC AND RUBBER COVERS

(Goodyear Tire and Rubber Co.)

Weight of Frictioned Fabric 1 Inch Wide, 1 Foot Long, 1 Ply Thick—Pounds			Weight of Rubber Cover 1 Inch Wide, 1 Foot Long
28-Ounce	32-Ounce	36-Ounce	$\frac{1}{32}$-Inch Thick
.026	.029	.032	.025
Weight per Square Foot. Pounds per Ply			Weight per Square Foot. $\frac{1}{32}$-Inch Thick
.312	.348	.384	.300

TABLE 4.—WEIGHT OF STANDARD RUBBER BELTS (28-OUNCE DUCK) POUNDS PER LINEAR FOOT

(B. F. Goodrich Rubber Co.)

For Heavier Ducks and Covers, See Notes Below

Ply of Belt	Thickness of Top Cover, Inches	Width of Belt, Inches								
		10	12	14	16	18	20	22	24	26
3	$\frac{1}{32}$	1.05	1.26	1.47	1.68	1.89				
3	$\frac{1}{16}$	1.25	1.50	1.75	2.00	2.25		Note 1		
3	$\frac{1}{8}$	1.65	1.98	2.31	2.64	2.97				
3	$\frac{3}{16}$	2.05	2.46	2.87	3.28	3.69				
4	$\frac{1}{32}$	1.28	1.54	1.79	2.05	2.32	2.56	2.82	3.07	3.32
4	$\frac{1}{16}$	1.48	1.78	2.07	2.37	2.66	2.96	3.26	3.55	3.87
4	$\frac{1}{8}$	1.88	2.26	2.63	3.02	3.39	3.76	4.14	4.52	4.90
4	$\frac{3}{16}$	2.28	2.74	3.20	3.65	4.10	4.56	5.02	5.47	5.92
5	$\frac{1}{32}$	......	......	......	2.40	2.70	3.00	3.30	3.60	3.90
5	$\frac{1}{16}$	......	......	......	2.72	3.06	3.40	3.74	4.08	4.42
5	$\frac{1}{8}$	......	......	......	3.36	3.78	4.20	4.62	5.04	5.46
5	$\frac{3}{16}$	......	......	......	4.00	4.50	5.00	5.50	6.00	6.50
6	$\frac{1}{32}$	......	......	......	......	......	3.44	3.78	4.13	4.47
6	$\frac{1}{16}$	......	......	......	......	......	3.84	4.22	4.61	5.00
6	$\frac{1}{8}$	......	......	......	......	......	4.64	5.21	5.57	6.03
6	$\frac{3}{16}$	......	Note 2	......	......	......	5.44	5.98	6.54	7.08
7	$\frac{1}{32}$	......	......	......	......	......	......	......	4.68	5.07
7	$\frac{1}{16}$	......	......	......	......	......	......	......	5.17	5.69
7	$\frac{1}{8}$	......	......	......	......	......	......	......	6.13	6.74
7	$\frac{3}{16}$	......	......	......	......	......	......	......	7.08	7.68

Ply of Belt	Thickness of Top Cover, Inches	Width of Belt, Inches								
		28	30	32	34	36	42	48	54	60
5	$\frac{1}{32}$	4.20	4.50	4.80	5.10	5.40				
5	$\frac{1}{16}$	4.76	5.10	5.44	5.78	6.12		Note 1		
5	$\frac{1}{8}$	5.88	6.30	6.72	7.14	7.56				
5	$\frac{3}{16}$	7.00	7.50	8.00	8.50	9.00				
6	$\frac{1}{32}$	4.82	5.06	5.51	5.85	6.19	7.22			
6	$\frac{1}{16}$	5.38	5.76	6.14	6.53	6.92	8.06			
6	$\frac{1}{8}$	6.50	6.96	7.43	7.88	8.35	9.75			
6	$\frac{3}{16}$	7.62	8.16	8.71	9.25	9.80	11.42			
7	$\frac{1}{32}$	5.46	5.85	6.24	6.73	7.02	8.20	9.36	10.52	11.70
7	$\frac{1}{16}$	6.03	6.46	6.88	7.32	7.75	9.04	10.32	11.60	12.90
7	$\frac{1}{8}$	7.14	7.66	8.17	8.67	9.18	10.80	12.23	13.76	15.30
7	$\frac{3}{16}$	8.27	8.86	9.45	10.03	10.62	12.38	14.15	15.92	17.70
8	$\frac{1}{32}$	6.08	6.53	6.96	7.40	7.82	9.14	10.40	11.70	13.00
8	$\frac{1}{16}$	6.65	7.13	7.60	8.08	8.55	9.96	11.38	12.78	14.22
8	$\frac{1}{8}$	7.77	8.32	8.87	9.42	9.97	11.63	13.30	14.95	16.62
8	$\frac{3}{16}$	8.88	9.53	10.13	10.78	11.40	13.31	15.21	17.12	19.00
9	$\frac{1}{32}$	......	......	......	8.16	8.64	10.08	11.52	12.96	14.40
9	$\frac{1}{16}$		Note 2		8.84	9.37	10.92	12.48	14.03	15.63
9	$\frac{1}{8}$	......	......	......	10.20	10.80	12.60	14.40	16.20	18.00
9	$\frac{3}{16}$	......	......	......	11.56	12.24	14.28	16.32	18.36	20.40

Note 1. Belts above the heavy line are too thin for troughing and for heavy loads.
Note 2. Belts below the heavy line are too thick to trough properly on standard idlers.
Note 3. For 32-ounce duck add .003 lb. per ply per inch of width.
Note 4. For 36-ounce duck add .006 lb. per ply per inch of width.
Note 5. For cover on pulley side $\frac{1}{16}$ inch thick instead of the usual $\frac{1}{32}$ inch add .025 lb. per inch of width.

covers and of the single plies of frictioned fabric made of standard conveyor belt ducks.

Table 4 (B. F. Goodrich Rubber Co.) gives weights per foot of finished belts with top covers of various thicknesses. The last lines show the additions necessary for ducks heavier than 28 ounce and for covers on the pulley side heavier than $\frac{1}{32}$ inch.

Weights of rubber belts of various makes are not quite the same, and are subject to some variation even in any one make. The figures given in the tables are, however, fairly representative of American belt manufacture.

Weights of canvas belts are given in Table 5, page 49.

Weights of balata belts of 38-ounce duck are given in Table 6, page 51.

Flanged Rubber Belts.—In the Edison ore concentrating plant (see p. 10) from 1893 to 1896 there was serious trouble from the lengthwise cracking of the belts due to the steep angle (45°) of the troughing idlers. When the Edison cement plant was built a few years later, no belts were troughed; all were run flat and some had light rubber flanges about $\frac{3}{4}$ inch high to retain the material. When these flanged belts ran through a tripper or over a reverse bend, there was a tendency to shear the edges off because the lower pulley of the tripper could not be made the full width of the belt, but had to clear the flanges of the belt. On the return run the flanges rubbed and chafed against the ends of the idler pulleys which were narrower than the belt; as a consequence, the flanges were destroyed. Most of these belts were later replaced by troughed belts.

If the flanges had been made stiff and heavy, the edges of the belt might have been supported at the lower pulley of the tripper and on the return run, but that construction is open to other objections and is costly.

Flanges (Fig. 51) are used on vanner belts employed in the wet concentration of metal-bearing ores. These belts are from 4 to 6 feet wide, made 2- or 3-ply with rubber flanges about $\frac{3}{4}$ or 1 inch high; they run over end pulleys only 12 or 14 inches in diameter and about 10 feet centers. The speed is slow, usually less than 5 feet per minute and the load is light, being merely a bed of water and ore about $\frac{1}{4}$ or $\frac{1}{2}$ inch deep. The stretch at the edge of a $\frac{3}{4}$-inch flange in going halfway round a 12-inch pulley is $4\frac{1}{4}$ inches in 18 inches, and the rubber must be of very good quality to withstand it. To strengthen the flange at the base, to prevent it from cracking there, and to prevent cracks from extending too far in from the top edge, vanner belt flanges have been reinforced by the insertion of a flexible cord of pure rubber, or an insertion of fabric, or by running the plies of fabric from the body of the belt up into the flanges. Church, in 1914, patented a vanner belt with flanges detachable for repairs.

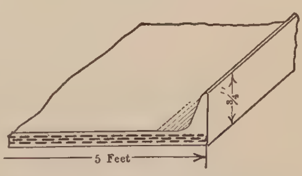


Fig. 51.—Flanged Edge of Vanner Belt used in Wet Concentration of Ores.

Flanged belts have also been used in ore-reduction plants to carry wet concentrates, the idea being that a troughed belt would not hold the wet

and semi-fluid material so well and would be apt to spill. In some of these, as in the Edison plant, the support of the return belt gave trouble even when separate grooved idlers were used at the ends of the return pulleys, to support the flanged edges. Conveyors of this kind have no real advantage and they are seldom used now; a troughed belt of the proper width and properly fed, works better than a flanged belt, and the cost for equal quality is considerably less, maintenance costs less and since the parts are not special, repairs can be made with less delay.

In a vanner belt the carrying surface must be flat by nature of the process used in concentrating the ore, and the flanges are necessary to retain the layer of ore and water; the flanges are always troublesome, however, and they should not be used on belts used solely to convey material.

Patents have been granted on a number of devices to increase the carrying capacity of a flat belt by turning up or flanging its edges, but none has come into commercial use. One of them (507156 of 1893) covers a belt with side strips fastened on by flexible connections so that while the strips normally lay in the same horizontal plane with the main belt, they could be turned up on the carrying run to form a trough-shaped section. Belts with sectional overlapping metal flanges have also been designed. Ridgway, in 1899, patented a belt with tubular edges which were intended to act as flanges to retain material on the belt.

Effect of Light, Heat and Age.—The absorption of oxygen by rubber causes what is known as drying or aging. It lessens the tensile strength and stretch of rubber and manifests itself by fine cracks in the surface of a belt. Since the chemical change goes on faster in sunlight, it is advisable to store rubber belts in a dark place and to protect them from direct sunlight when in use. At a mine in Arizona a belt working in a conveyor gallery was considered worn out and was replaced by a spare belt of the same make which through carelessness had lain exposed to the sun for some months. The spare belt lasted only a few weeks; it was removed and the old worn belt was put on until a new belt could be obtained.

Belts age more rapidly when they are exposed to heat in storage or when they carry hot materials.

Figs. 52, 53 show the effect of age on the tensile strength and stretch of various rubber compounds (not belts). Other tests reported by the Bureau of Standards (Bulletin 38, edition of 1921) indicate that compounds which tested from 1000 to 2600 pounds per square inch when fresh lost 20 to 30 per cent in strength after three years' storage and 40 to 60 per cent after six years' storage in a dark, well-ventilated place at 75° F. The tests also show that while the nature of the compounding ingredients has some effect on aging, the effect of "over-cure" or "under-cure," especially the former, is much more marked. These terms refer to the longer or shorter time during which the rubber is held at the vulcanizing heat.

Much of the improvement in rubber-belt making in recent years has come as a result of a better knowledge of what to do to lessen the effects of heat, light and age.

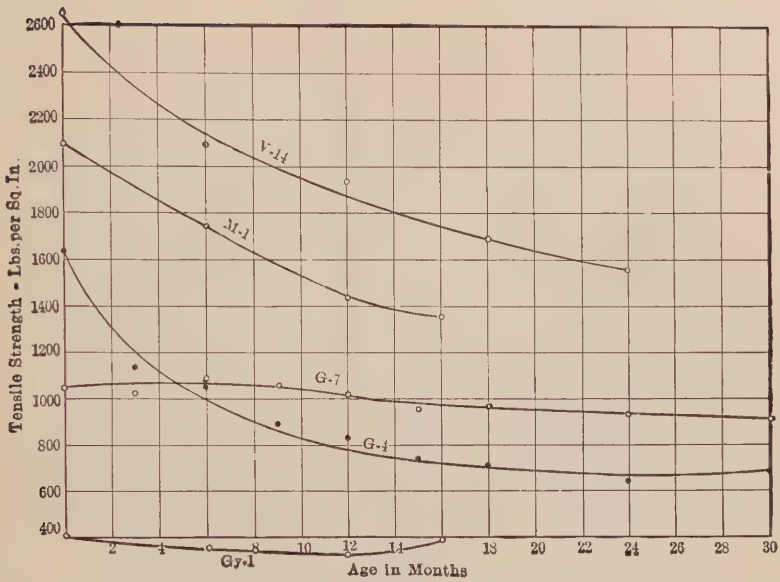


FIG. 52.—Effect of Age on Tensile Strength of Five Specimens of Rubber Compounds. (Bureau of Standards, Washington, D. C.)

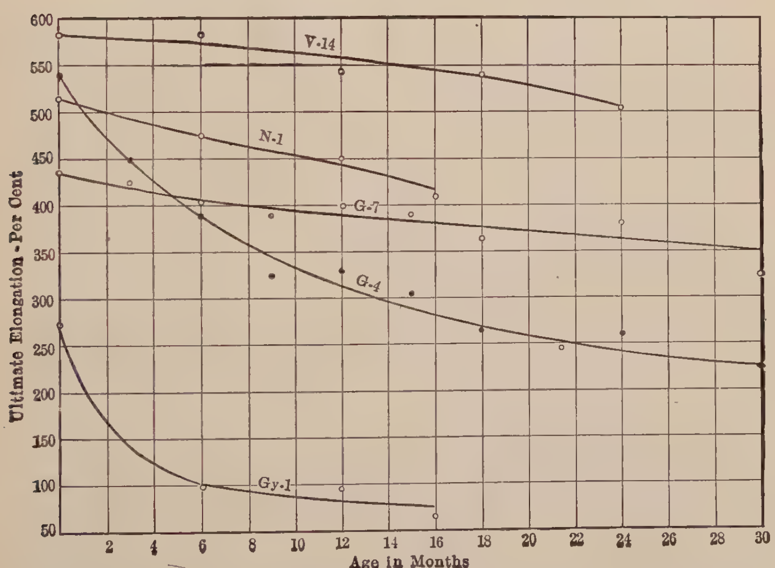


FIG. 53.—Effect of Age on Stretch of Five Specimens of Rubber Compounds. (Bureau of Standards, Washington, D. C.)

How to Keep Rubber Goods.—The following information is furnished by the B. F. Goodrich Rubber Co:

"No matter how good the quality of rubber, or how short a time it is to be kept for use, it is worth while to give some attention to storage conditions. The oxidation of rubber is promoted by the presence of heat, light and air. Ordinarily, room temperatures are satisfactory except close to the ceiling or in a room under the roof. Basement rooms are preferable, and the ideal storage place is a cool, dark cellar protected from frost in winter and heat in summer, yet not too close to the heating apparatus. Dry air deteriorates rubber much more rapidly than moist air, though an excess of moisture is not best for articles like belts which are made partly of fabric. A cellar storeroom is better if the floor or walls are natural earth, provided there is sufficient ventilation to avoid excessive moisture.

"Since the air in artificially heated buildings in winter is usually excessively dry, it is advisable, where practicable, to provide some special means to maintain normal humidity in the storeroom. Some users of rubber goods provide humidors in the form of closed bins or boxes in which are kept bricks previously soaked in water, or flat trays kept filled with water.

"In cases where it is not convenient to provide special storage arrangements, care should at least be exercised to avoid placing rubber belts and other rubber goods near steam pipes, radiators, hot-air registers, windows or ceilings."

CANVAS BELTS

Stitched canvas belts used for elevators and conveyors are generally made of cotton duck that weighs 32 ounces per square yard. It is more closely woven and heavier than 32-ounce duck used for rubber belts because the trade custom is to rate the latter by the weight of a piece 42 inches wide, 1 yard long; on that same basis, 32-ounce canvas belt duck would be rated as 37-ounce.

There are various weaves of 32-ounce duck, depending on the service of the belt and the experience of the belt-maker. In general, the warp threads are closer, say, 28 per inch, than the filler threads, which may be 16 per inch. Sometimes the threads are all of the same size, but some belt-makers prefer to have the filler threads of conveyor belts heavier than the warp threads because they take most of the abrasion, and also to make a stronger joint, when, as often happens, the belt fastener used is of the wrong type or is improperly applied. For fasteners see p. 56.

Fig. 54 is a diagram showing, in a conventional way, the construction of a 6-ply stitched belt for elevator or conveyor service. Duck of the proper width is folded and assembled to make the 4 inside plies and these are stitched together by rows about 1 inch apart. Then the wrapper or cover is added to make 2 more plies and the whole belt is sewed again; this time the stitches are in rows $\frac{1}{4}$ inch apart. The thread is a strong cotton twine, heavier than the warp or filler; the lockstitch between the needle thread and the bobbin thread of the sewing-machine is buried

within the thickness of the belt (see Fig. 54) and it is not likely to come loose even when the threads are worn away on either side of the belt. In that respect the sewing is like that which fastens on the sole of a shoe; the sole is held on even after the exposed stitches have been worn away.

The next process with some makers is to subject the assembled plies to a combination of pressing and stretching, after which the belt is immersed in the waterproofing compound. It is then run between rolls to squeeze out the excess of liquid. Some makers stretch the belt during this operation, but in all factories the wet belts are finally dried and " cured " by stretching them out in horizontal spans 150 feet or more in length and keeping them under tension for some days or weeks to season. During this time the oil in the compound dries or partially oxidizes to a gum; when linseed oil is used, the gum formed has less strength and less elasticity than rubber, but it is proportionately less affected by light, heat and age. To a certain extent, it acts like the oxidized linseed oil which forms the basis of floor linoleum, and when properly applied and " cured " it gives a belt much greater resistance, to abrasion than the raw cotton possesses. Belts saturated with mineral oils, semi-drying oils and other substitutes for

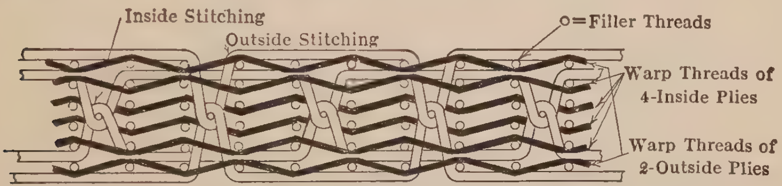


FIG. 54.—Diagram Showing Assembly of a 6-ply Stitched Canvas Belt.

linseed oil are more flexible, but they do not resist wear so well and are more likely to stretch in service. Belts that are not maintained under stretch in the factory long enough to dry or set the oil properly will also show excessive stretch and a shorter life. The thicker the belt, the longer it should be cured; a good 8-ply belt may be kept under stretch for several months before it is considered ready to use. The output of a belt factory can be increased and the cost of the product decreased by shortening the time of " cure," but it is apt to be at the expense of quality in the belt.

Saturating compounds for canvas belts are of various kinds depending on the work of the belt.

Class 1.—Drying Compounds. Belts for ordinary elevator and conveyor service are generally soaked in a thin oil compound, the basis of which is linseed oil. It is thin enough to penetrate all the fibers of cotton in the belt and after the excess has been squeezed out, a definite quantity of it remains in the fibers to waterproof them to a certain degree, to lessen the wear due to internal bending and friction and when properly " cured " to make the threads of warp and filler tougher and more resistant to abrasion. Such belts are then coated with a paint made of drying oil and a mineral pigment; this fills up the stitching holes and when dry, gives the stitching

twine and the outside plies of the belt a tough hard surface. This surface coat also gives the belt a good coefficient of friction for contact with the pulley and it prevents the saturating liquid remaining in the cotton fibers from drying out during the life of the belt. Belts treated with Class 1 compounds are not suitable for working in the wet, or under great heat, nor where they are subject to acid or alkali dust or fumes.

Class 2.—What are known as asphalt compounds are mixtures of asphalt or gilsonite with various oils and gums. They make a belt that withstands water, heat and chemical action, but one which does not resist surface wear so well as those treated with Class 1 compounds. Transmission belts for high speed work on small pulleys are saturated with compounds of this class.

Class 3.—Colorless waterproofing compounds are used for belts that convey bread, crackers and other food products, wrapped packages in stores, books, fabrics, etc., which must not be discolored in handling. Compounds of Class 1 and Class 2 are open to objection in this respect. Class 3 compounds leave the canvas its natural color and give sufficient protection against atmospheric moisture; they do not give a belt great resistance to the rough handling of heavy packages nor to the actual contact of wet material.

Class 4.—Tasteless, odorless waterproofing compounds are put into belts used in canneries to handle fruit and vegetables. They contain a wax which protects the cotton against the action of fruit acids, and being light in color, they give the belt that appearance of cleanliness which is desired in canneries. The compound is not affected by hot water or steam which is used in cleaning such belts to keep them sanitary.

Stitched Canvas Belts for Conveyors.—Canvas belts for handling bulk materials and heavy packages are generally treated with a Class 1 compound (see p. 47), and when properly made they stretch no more than a rubber belt and possess considerable resistance to cutting and abrasion. This resistance is less than that of a belt with a rubber cover, but the canvas belt is homogeneous while the rubber belt is not. The cover and the friction rubber in a rubber belt protect the raw cotton duck from wear, and when the wear has gone through the cover and into the duck, the plies are held together only by the tenacity of the friction gum. This rubber deteriorates on exposure to the air and the raw cotton is liable to decay from the action of moisture and dirt. In a properly treated fabric belt, the whole thickness is equally resistant to abrasion, the plies are held together by stitching, which is to a certain extent effective even when the outer plies are worn away, and the threads and fibers of the duck do not mildew from the absorption of moisture, as raw cotton does. When the cover of a rubber belt and its first friction layer are worn away, it approaches the end of its life, but a canvas belt is intended to expose its whole thickness to wear, and it should bear the loss of several of its plies before it is discarded.

Protective Mineral Coating.—It has been suggested that the surface of canvas belts be protected from abrasion by a coating composed of a

mixture of stone with tar or asphalt. The difficulties are that if the coating is soft, it will stick to the return idlers, while if it is hard, it will crack and fall off. Canvas belts carrying cement, fine crushed stone, etc., are sometimes treated with a thin coat of non-drying paint or asphaltic material. Some of the fine stuff in the material carried becomes attached to the carrying surface and in certain cases it forms a protective layer or skin which increases the life of the belt.

Flexibility of Canvas Belts.—A canvas belt can be made as flexible for troughing as a rubber belt of the same number of plies by using duck with a light filler or by treating it with a non-drying compound, or by using it while still soft and unseasoned; but such a belt will not last long under hard work. To resist abrasion well, a belt should have a strong filler and the saturating compound should be of Class 1 with a high percentage of linseed oil. When properly cured it will have a density and a toughness that are a measure of its wearing quality. It will not be as flexible as a rubber belt and will not bear troughing so well, especially in the narrow widths.

For weights of stitched canvas belts see Table 5.

Balata belts are made of cotton duck waterproofed and held together by balata, a tree-gum brought from the West Indies and the north coast

TABLE 5.—WEIGHT OF OILED AND PAINTED STITCHED CANVAS BELTS

(Main Belting Co.)

For Asphalt-treated Belts Add 10 Per Cent

Width, Inches	Weight of 1 Foot of Belt, Pounds						
	4-Ply	5-Ply	6-Ply	7-Ply	8-Ply	10-Ply	12-Ply
12	1.30	1.63	1.95				
14	1.52	1.90	2.28				
16	1.73	2.17	2.60	3.04			
18	1.95	2.44	2.93	3.42			
20	2.17	2.71	3.25	3.79			
22	2.39	2.98	3.58	4.17	4.77		
24	2.60	3.25	3.90	4.55	5.20		
26	2.82	3.52	4.23	4.93	5.64	7.05	
28	3.04	3.79	4.55	5.31	6.07	7.59	
30	3.25	4.07	4.88	5.69	6.50	8.13	
32	3.47	4.34	5.20	6.07	6.94	8.67	
34	3.69	4.61	5.53	6.45	7.37	9.21	
36	3.90	4.88	5.85	6.83	7.81	9.76	11.71
38	4.12	5.15	6.18	7.21	8.24	10.30	12.36
40	4.34	5.42	6.50	7.59	8.67	10.84	13.01
42	.....	5.69	6.83	7.79	9.11	11.38	13.66
44	.....	5.96	7.15	8.35	9.54	11.92	14.31
46	.....	6.23	7.48	8.73	9.97	12.47	14.96
48	.....	6.50	7.81	9.11	10.41	13.01	15.61

NOTE.—Belts between Heavy Lines are well suited to idlers shown in Fig. 67.

of South America. The gum is plastic at 212° F., stronger than rubber at ordinary temperatures, but not so elastic. It does not absorb oxygen from the air to the extent that rubber does; hence it retains its life and strength for a long time.

The raw gum is washed, as rubber is, to remove dirt, then dried, cut up and dissolved in a liquid solvent which when applied to the duck, carries the gum into the threads and fibers of the cotton. The duck is generally of a closer weave than that used for rubber belts; in some balata belts it weighs 28 to 32 ounces per square yard, but as made for conveyor and elevator service, the duck in the best balata belts weighs 38 to 40 ounces and has the warp (lengthwise) threads somewhat heavier than the filler (crosswise) threads.

In making the belt, the duck is washed to get rid of any oil or sizing used in spinning or weaving the yarn; then it is dried, run through the solution of gum and dried again. After cutting to the right width, it is folded, refolded, and rolled under pressure while warm to build up the required thickness. It is then stretched and allowed to cool under strain. This makes a dense, strong belt, rather stiff, but not likely to stretch, and since its fibers are impregnated with the gum, it shows a high resistance to the absorption of water (see Table 10). The gum softens at 120° F. and the belt must not be used where it may get too hot.

Balata belts have been made in Great Britain with rubber covers fastened on by cement and by rows of copper-wire stitching, but that style is not known in this country. They have been made here with a plastic layer of gutta-percha gum, which is similar to balata, rolled on to the wearing surface, but it is not so resistant to abrasion as rubber is, and cannot be made so thick. Balata belts have been used for conveyor work to a greater extent in Europe than in this country; this may be attributed chiefly to the fact that the manufacture of rubber belts has received more attention in this country than anywhere else and competition from other kinds of belt has been made more difficult here than abroad. In this country balata belts compete with others more actively as transmission belts and elevator belts, in the latter service because of their great strength and freedom from stretch, and especially in elevating wet materials like mineral pulps where the waterproof quality of the balata belt is an advantage. In conveyor service they do not trough so well as rubber belts because of their density and stiffness; hence they are better suited to run on shallow troughing idlers or flat rollers rather than on three-pulley or five-pulley idlers troughed for 30°. In this respect they are similar to canvas belts with Class 1 impregnation.

For weights of balata belts see Table 6.

Solid-woven Cotton Belts.—Unlike other belts, these are not woven as duck, a ply at a time, but are woven in their full thickness in looms built for the purpose. Layers of warp (lengthwise) threads under tension are woven in with layers of filler (crosswise) threads to make the required thickness and the whole mass is bound together in the loom by lines of

TABLE 6.—WEIGHT OF BALATA BELT (38-OUNCE DUCK)

(R. & J. Dick Co.)

Width, Inches	Weight of 1 Foot of Belt, Pounds						
	3-Ply	4-Ply	5-Ply	6-Ply	7-Ply	8-Ply	9-Ply
12	.83	1.11	1.40				
14	.97	1.30	1.63				
16	1.11	1.48	1.85	2.22			
18	1.25	1.67	2.09	2.52			
20	1.39	1.85	2.31	2.79	3.24		
22	1.53	2.04	2.54	3.06	3.56		
24	1.67	2.22	2.76	3.33	3.88	4.43	
26	1.80	2.40	3.00	3.61	4.20	4.80	5.40
28	1.94	2.59	3.23	3.88	4.52	5.17	5.81
30	2.08	2.77	3.46	4.16	4.84	5.54	6.23
32	2.22	2.96	3.69	4.43	5.17	5.91	6.64
34	2.36	3.14	3.92	4.71	5.49	6.28	7.06
36	2.50	3.33	4.15	4.99	5.81	6.65	7.47
38	2.64	3.51	4.38	5.26	6.14	7.02	7.89
40	2.78	3.70	4.61	5.54	6.47	7.39	8.31
42	2.91	3.88	4.85	5.82	6.79	7.76	8.73

binder threads which pass around the filler threads from face to back of the belt. Fig. 55 shows, in a diagrammatic way, sections lengthwise and crosswise of such a belt. The binder threads serve the same purpose as the

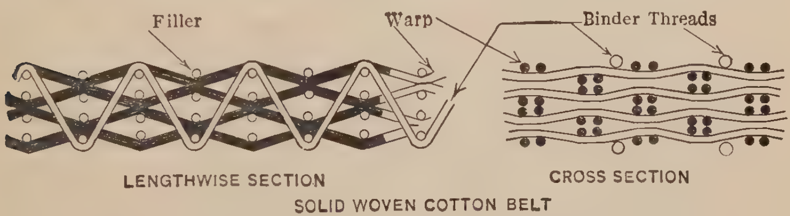


FIG. 55.—Diagram Showing Assembly of a Solid-woven Belt.

stitching used in a canvas belt (see Fig. 54); without them the belt will not hold together.

Solid-woven belts are not sold on specifications as to size of threads, weight of assembled belt or kind of weave. They are known by various trade names and the thicknesses are designated by terms which are somewhat arbitrary. Those belts used for conveying and elevating service have generally 6 or 8 layers of warp threads and for tensile strength correspond to 6- or 8-ply canvas or rubber belt.

Table 7 and Table 8 give facts about two makes of solid-woven belt.

Solid-woven belts were made originally for power transmission and for that purpose they are oftenest used. When impregnated they are treated with a Class 2 waterproofing compound (see p. 48) and are cured by

TABLE 7.—"WOOSTER" SOLID-WOVEN COTTON BELTING

(Duryea Mfg. Co., Bayonne, N. J.)

Kind of Belt	Average Strength, Pounds per Inch Width	Average Weight per 1 Inch Wide per 100 Ft., Pounds	Average Thickness, Inch	Equivalent in Stitched Canvas or Rubber Belt
Light	800	11	$\frac{3}{16}$ to $\frac{7}{32}$	4-ply
Medium	1400	14	$\frac{1}{4}$ to $\frac{9}{32}$	6-ply
Heavy	2300	17	$\frac{5}{16}$ to $\frac{3}{8}$	8-ply

TABLE 8.—"SCANDINAVIA" SOLID-WOVEN COTTON BELTING

(Scandinavia Belting Co., New York)

Kind of Belt	Average Strength, Pounds per Inch Width	Average Weight per 1 Inch wide per 100 Ft., Pounds	Average Thickness, Inch	Equivalent in Stitched Canvas or Rubber Belt
Single	1300	10	$\frac{3}{16}$	4
Extra stout	2300	15	$\frac{5}{16}$	6 or 7
Triple	3000	20	$\frac{7}{16}$	8 or more

stretching them out in long spans in the way stitched canvas belts are treated. As they come from the loom they are more flexible than stitched canvas belts, and with a Class 2 impregnation, they retain that characteristic and hence conform very well to the contour of standard troughing idlers. They do, however, lack the hard surface of painted canvas belts and are not so tough as those treated with Class 1 compounds; and in many of them the binder threads are lighter than the stitching threads used in canvas belts. They are apt to stretch more in service than rubber belts and canvas belts, but they have a high coefficient of friction for contact with the driving pulleys.

Solid woven belts are frequently used without waterproofing for light package-conveyors in stores where they are always under cover and not subject to changes of temperature nor to severe pulls. Table 9 gives a comparison between two specimens of such belting and one stitched canvas belt as to stretch and ultimate strength.

"R. F. & C." (rubber filled and covered) belts are made by the Buffalo Weaving and Belting Co., Buffalo, N. Y. They are solid-woven cotton belts impregnated with a rubber solution and enclosed in a rubber cover. For conveyor work the top cover is made thicker than the cover on the pulley side of the belt. By laboratory test they are more nearly waterproof than other kinds of belt (see p. 56).

Strength of Belts.—In a rubber, canvas or balata belt the ultimate strength depends largely on the strength of the duck of which it is made; in a solid-woven belt the strength varies according to the sizes of the threads and the closeness of the weave. Belt ducks are generally referred to as weighing so much per square yard, or per yard of length 42 inches wide,

TABLE 9.—TESTS OF SEVERAL BELTS OF KINDS USED FOR PACKAGE CONVEYORS

Total Load, Pounds	Per Cent Stretch in $6\frac{1}{4}$ Inches		
	Solid Woven 6″ × 4-Ply		Stitched Canvas 6″ × 4-Ply
	No. 1	No. 2	
600	8.4	7.3	1.9
1200	11.3	10.1	2.7
1800	12.6	12.3	3.4
2400	14.3	13.6	4.3
3000	15.6	14.8	5.1
3600	17.0	16.4	5.8
4200	18.2	17.3	6.9
4800	18.7	18.2	7.6
5400	Broke at 5330 lbs.	Broke at 5230 lbs.	8.9
6000	....	....	9.5
6600	....	....	10.8
7200	....	....	Broke
Breaking strength in pounds per inch width per ply.....	222	218	300

These tests were made in a 50,000-pound Riehle Machine. The movement of the jaws was not stopped from the time the machine started until the belt broke. The record of stretch was taken electrically at the increments of load indicated in the left column. Belt specimens were 18 inches long, 10 inches clear between jaws of machine.

and in a general way, the heavier the duck, the stronger it is. There are, however, many different ways of assembling warp threads and filler threads to make a duck weigh so many ounces per square yard; a duck for a canvas belt may have a filler relatively heavier than in the same weight of duck made for a rubber belt. The duck for a balata belt can be more closely woven than a rubber belt duck. It is therefore not possible to compare the strengths of belts solely on the basis of the weights of the duck.

For reasons stated on page 39, there is no direct connection between the strength of a finished belt and the strength of the duck as tested by the strip method or the grab method. The plies do not all take an equal share of the load, nor is the load uniformly distributed to all of the warp threads in the width of the belt. The "friction" or the impregnation of a belt has an important influence on its strength. In a rubber belt, the layers of "friction" rubber act as a support for the threads, prevent distortion of belt structure under load and help to distribute the load among the plies of duck. In a stitched canvas belt, the oxidized or gelatinized oil which fills the spaces among the threads acts in a similar way, but with less effect, because it is not so strong as the rubber. In a balata belt, the saturating gum is stiff and strong, and with a given structure of duck it makes a belt higher in tensile strength and less in stretch than with other means of holding the plies together. In the best impregnated solid-woven belts, the structure

of fabric is quite dense and the strength is high (see Tables 7 and 8), but in most plain, white, solid-woven belts the weave is not so close, the threads are not kept in place by an impregnating gum and consequently the stretch is greater and the breaking strength is less. Table 9 records tests of two solid-woven belts not impregnated and one stitched canvas belt with Class 1 impregnation.

Strength of Rubber Belts.—Since no two makes of rubber belts are alike as to the weave of the duck even for the same nominal weight, it is not possible to assign definite strengths to the belts, considering also the contingencies of manufacture. It is really not necessary that belts should be rated by their breaking strengths; as is pointed out in Chapter V belts are seldom strained to more than one-twelfth or one-fifteenth of their ultimate strength, and with most belts, the factor which determines their suitability to particular service and their life in service is the amount of stretch, rather than the breaking strength.

As a guide for determining the allowable working tensions, the following may be taken as representing the average breaking strengths of rubber belts as made in this country, measured per inch of width per ply of thickness:

Belts made of 28-oz. duck	300 lbs.
Belts made of 30- or 32-oz. duck	325 lbs.
Belts made of 36-oz. duck	360 lbs.

That is, a

20-inch 5-ply belt of 28-oz. duck is good for about 30,000 lbs.
36-inch 6-ply belt of 36-oz. duck is good for about 77,000 lbs.

Strength of Fabric Belts.—Stitched canvas belts of 32-ounce duck will break at about 300 pounds per inch per ply.

Balata belts are made of various weights of duck; tests of the best grade using 38- or 40-ounce duck show about 400 pounds per inch per ply. In general they are about 20 or 25 per cent stronger than rubber belts of 28- or 32-ounce duck.

On the relation between ultimate strength and working tensions see Chapter V.

For weights of belts see Tables 3, 4, 5, 6, 7, 8.

Various Belts for Different Kinds of Service.—American practice in belt-conveying has for years been close to the rubber belt business, and more rubber belts are used for conveyor work than belts of other kinds; stitched canvas, balata or solid-woven cotton belts. Rubber belts can be made to handle economically material of all kinds, light or heavy, fine or coarse, wet or dry. For carrying hard material in heavy pieces a high-grade rubber belt with a good cover should make the best conveying medium, but it would not be economy to use such an expensive belt for carrying crushed coal, and it would be a waste of money to use it on a package conveyor. A rubber belt of medium grade with a light cover will in most cases carry small coal for less cost per ton or per year than a high-grade

belt, especially since the life of a belt so often depends upon factors external to the belt itself (see Table 32, page 197).

In conveyors where there is no severe cutting action from the impact of material at the loading point, and where the belt is protected from the weather, it is often possible to use canvas belts with economy. The relative prices of canvas belts and rubber belts depend largely on the costs of raw cotton and raw rubber. Since these fluctuate, the ratios of prices are not constant, but, in general, canvas belts cost less than rubber belts of equal width and ply. No one thinks that a canvas belt resists cutting and abrasion so well as a rubber belt with a cover, but when its lower cost is considered, it may be economy to use it and get less tonnage or a shorter life than to pay much more for a rubber belt. On the other hand, buying canvas belts for some conveyors would be throwing money away; the material may be too heavy or too sharp, the conditions of loading and discharge, lubrication, care and general oversight may be so good that a rubber belt will have a chance to show a life much greater in proportion to its cost than any other kind of belt. In other words, if a canvas belt costs $750 and under the operating conditions lasts one year, it is better to pay $1000 for a rubber belt if it can be depended upon to last more than sixteen months; but if the average life of the rubber belts on the conveyor is no more than fourteen months, the canvas belt is more economical.

Canvas belts are sold under trade names that tell nothing about the make-up of the duck, the stitching, the nature of the waterproofing compound, the amount of stretch taken out and the time of seasoning or curing. Correct knowledge on these points is not widely disseminated. Some of the failures of canvas belts in service must be attributed to faulty methods of manufacture, but disappointments in service are sometimes due to using the wrong kind of belt. A canvas belt bought from a jobber's stock and run over troughing idlers may be a failure as a conveyor belt, although it might have been a good transmission belt. Conveyors are not all alike, and a belt suited to one may last only a short time on another; good elevator belts are not necessarily good conveyor belts. Canvas belts are naturally denser and stiffer than rubber belts and do not trough so well on some idlers which have been designed for use with rubber belts.

Canvas belts are not recommended for the difficult work of conveying or elevating sharp ore in the presence of water. Oil-treated belts with Class 1 saturation are not sufficiently waterproof. Belts treated with Class 2 compounds are better in that respect, but for continual stretching and bending while exposed to water, as in a wet elevator handling mineral pulps, they do not resist the water so well as a good rubber belt or a balata belt. They are also deficient in resistance to cutting and abrasion in the presence of water as compared with other belts.

Much of what has been said in the preceding paragraphs about canvas belts applies to balata belts also. They are stronger and stiffer than rubber belts and do not trough well on multiple-pulley idlers. They

have been used to advantage in handling wet sand and ores in conveyors equipped with cylindrical or flared idlers (see p. 50).

For an instance of a special use of canvas or balata belts see p. 190.

Absorption of Water by Various Belts.—Table 10 gives results of laboratory tests of the absorptive capacity of various kinds of belts. The specimens were all 4-inch 4-ply belts cut 5 inches long and treated alike. The figures in the table may not show which kind of belt absorbs the least moisture in actual service when subjected to stretching and bending in the presence of water; under those conditions and after a period of service, the belt becomes more pliable, the body becomes more open, and the percentage of absorption will be greater with belts of all kinds. Nevertheless the greatest absorption in service will be shown by those belts which give a high percentage in the laboratory tests and to that extent the figures of the table are a useful guide.

TABLE 10.—ABSORPTION OF WATER BY VARIOUS KINDS OF BELT

Four-inch, 4-Ply Belts, 5 inches long, Soaked 50 Hours in Water at 75° F.

	Per cent of Weight of Belt
Various rubber belts	4 to 10
Standard grade stitched and painted canvas belts	9 to 11
Cheaper grade stitched and painted canvas belts	12 to 22
Belts saturated with asphalt compounds	3 to 4
Untreated cotton belts	30 to 40
Balata belts	5 to 9
Rubber-filled and covered belt	2.7 (1 specimen)

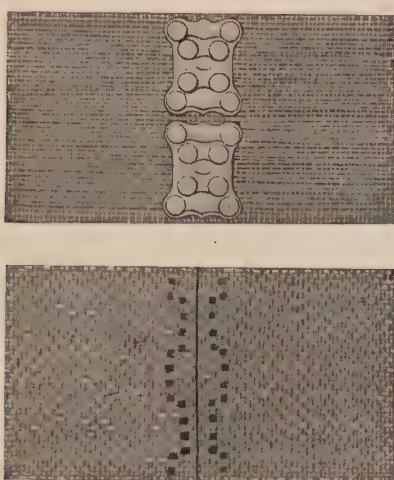
Belt Fasteners.—A fastener for a belt joint must be strong and yet so flexible or so short that it will bend around the pulleys without breaking the belt by bending it crosswise or pull apart by tearing out the cut ends of the belt. A fastener for a conveyor belt must also be flexible crosswise or be applied in short sections so that the belt can conform to the contour of the troughing idlers.

Rawhide lacing and the coiled wire lacing used for transmission belts are not used for conveyor belts. They do not make a closed joint and as applied to fabric belts they are objectionable because they transmit the pull to the filler or crosswise threads in the belt, and these are apt to pull out when the lacing holes are pierced close to the cut ends of the belt. To avoid pulling out and to transmit the pull to the warp or lengthwise threads, a fastener for a fabric belt should grip the warp threads by squeezing them together and preferably by clinching them in some way.

Fasteners for Conveyor Belts.—There are several styles of metal fasteners for fabric belts used for conveyors:

1. Steel plates with split rivets (Fig. 56) are made by the Conveying Weigher Co., N. Y.; The Bristol Co., Waterbury, Conn.; Crescent Belt Fastener Co., N. Y., and others. Except in very thick belts, it is not necessary to punch holes for the rivets; the rivets, when driven through the belt, compress the warp threads and clinch around them.

2. Steel clinch hooks are simpler, consisting only of a steel plate with projecting prongs which are driven through the belt by blows of a hammer

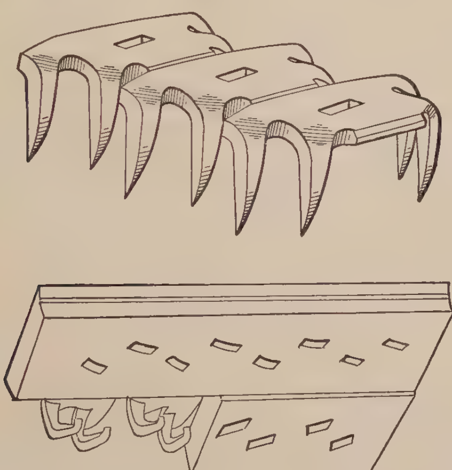


Upper Side of Belt Showing Steel Plates and Heads of Rivets.

Pulley Side of Belt Showing Ends of Split Rivets Spread and Imbedded in Belt.

Fig. 56.—Belt Joint with Steel Plates and Split Rivets.

and then clinched on the pulley side. In the "Turtle" fastener (Fig. 57) (Flexible Steel Lacing Co., Chicago) the prongs are set edgewise to the pull; they are tapered to compress the warp threads, drive easily and clinch



Before Inserting.

View of Pulley Side of Belt Showing Prongs Bent Over and Clinched.

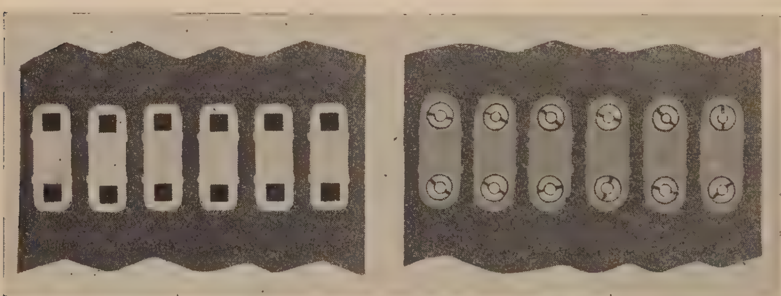
Fig. 57.—"Turtle" Belt Fastener.

on the pulley side of the belt. The Bristol fastener (Fig. 58) is similar in construction; the tapered prongs enter between the warp threads and get a firm hold by squeezing them together.

3. Bolted Fasteners are steel plates with bolts. The High Duty Belt Fastener (Flexible Steel Lacing Co., Chicago) (Fig. 59) is used only for belts $\frac{3}{8}$ inch or more in thickness that run over pulleys at least 2 feet in diameter. It does not leave the belt flush on either side and hence there is noise when it passes over idlers, but it is a strong fastener for very heavy belts. It requires the belt to be drilled for the bolts, but these take no shear; the hold depends upon the powerful compression between the opposite plates and the wedging action of the conical nuts. The Jackson fastener is similar in principle, but it has oval cup washers on the pulley side of the belt, and that side of the joint is smooth and runs quietly over the idlers. It is more fully described as a fastener for elevator belts. (See p. 264 and Fig. 248.)

Fig. 58.—Bristol Steel Belt Fastener.

4. Hinge-pin Fasteners.—" Alligator " lacing (Flexible Steel Lacing Co.) consists of sections of sheet steel with prongs driven into the belt from both faces and made to interlock over a steel or rawhide pin to form a hinge between the butt ends of the belt. It is well suited to belts run flat and for package conveyors where a close butt-joint is not essential. The fastener is thin, lies close to the belt and is not apt to damage packages at loading or discharge stations.



Upper Side of Belt Showing Steel Plates and Conical Nuts.

Pulley Side of Belt Showing Steel Plates with Square Heads of Bolts Set Flush.

Fig. 59.—High-duty Belt Fastener for Heavy Belts.

Sizes of Fasteners.—In using any metal fastener, be guided by the maker's instructions as to the way to apply it and the proper size to use for the particular width and thickness of belt. Some kinds suitable for leather belts make a poor joint in a fabric belt, the prongs or rivets are too close to the cut ends of the belt and the threads pull out. In using steel plates with split rivets, the rivets must be of the proper length for the thickness of belt.

A joint made by a metal fastener is never as strong as the body of the belt, and in a fabric belt it is a point of weakness because it opens the way for the entrance of moisture. In important work, the cut ends of the belt are covered with several coats of rubber cement before the fasteners are put on. This hinders water from getting into the belt.

Step-splices.—The construction of a step-splice is shown in Fig. 60. Allow a step of 3 or 4 inches for each ply of belt. Cut each ply carefully and avoid cutting into the ply below it. Coat the cut surfaces with rubber cement, allow it to dry. Put on a second coat, allow it to dry. Put on a third coat and when it is nearly dry but still tacky, press the two ends of the belt together and roll or beat the cemented surfaces into thorough contact. When the joint is dry, rivet or stitch the lapped ends together.

Vulcanized Joints.—Rubber transmission belts are sometimes made endless at the factory by splicing the ends and vulcanizing the joint; but conveyor and elevator belts are seldom, if ever, made in that way, because of the difficulty of getting an endless belt into place over the pulleys and idlers. Rubber belts for conveyors have occasionally been made endless,

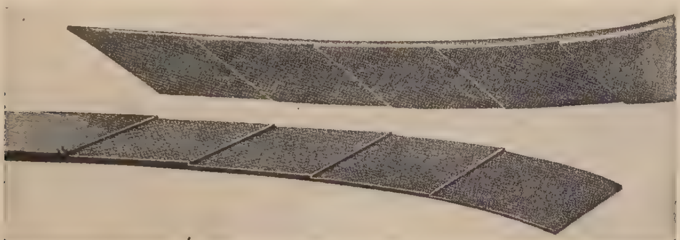


FIG. 60.—Ends of a 5-ply Belt Prepared for a Step Splice.

after being put in place, by the use of a portable steam-heated vulcanizing clamp. In this method, a stepped splice is coated with a vulcanizing rubber cement, then squeezed and heated for some minutes. When well done it makes a strong, tight joint that runs quietly over pulleys and idlers without clicks and bumps. It does, however, require skill and care to do the vulcanizing right and prevent the formation of blisters from the evaporation of moisture in the belt near the clamp.

The expense of the vulcanizing equipment and the necessity for skilled labor will prevent this method from coming into general use, but these factors do not weigh so heavily in the case of an important installation that uses many belts, especially if the belts are thick and hard-worked. Field vulcanizing equipments for such installations are not yet made in a commercial way, but it is probable that they will be developed to that point as thick and heavy belts come into wider and more general use.

Steel Belts.—Flexible steel bands are used in Europe as conveyor belts; there are said to be over 1000 installations in Germany, Sweden and England (1922). Very few have been built in the United States. In 1911 one was installed at a smelter in Utah; it was 24 inches wide, of No. 14

gage soft steel (0.08 inch thick) and ran 100 feet per minute over cylindrical rollers on the upper run; it slid back on the empty run. The conveyor was 50-foot centers, horizontal, and carried copper matte shoveled from railroad cars. The end pulleys were 48 inches diameter, cast iron, not lagged. The conveyor was used a few months, then scrapped. The trouble was in the repeated breakage of the steel band, probably from the blows from the matte shoveled onto the band and from material caught between the return belt and the foot pulley. Besides, soft steel is not elastic enough to withstand continual bending back and forth over pulleys and idlers.

A similar steel belt, somewhat wider, was used in a foundry in the Pittsburgh district about fifteen years ago to carry molds on the upper run and empty flasks on the return. It was not a success.

Sandvik Belts.—This belt, made at Sandvik, Sweden, is a thin band of cold rolled steel 0.03 or 0.04 inch thick, very flexible, but not capable of troughing. It comes in lengths up to 300 feet and in widths up to 16 inches; the makers are not in position to roll wider bands.

European practice is to use end pulleys not less than 40 inches in diameter and lag them with fabric belt or rubber belt to maintain driving contact, and especially to prevent the fine particles which adhere to the band from being jammed against the pulley rim and causing dents, buckles or breaks in the band. The speed is never over 300 feet per minute, usually less than 200 feet. For short runs and for materials not especially abrasive, the carrying run slides in a trough like the old-time grain conveyors (Fig. 20). The empty run may slide back or return over idlers; these may be 25 to 35 feet apart if clearance for the sag permits. When the loaded run cannot be slid, it is carried on flat-faced rollers (Fig. 61), 12 to 18 inches in diameter. Especial care must be taken to keep pieces of material from falling on the return belt and getting between it and the end pulley, or the thin steel ribbon is apt to be buckled or broken. The same thing may happen if a hard crust of material forms on the rims of the pulleys. When discharge is not over the end pulley, a diagonal plow or scraper (Fig. 149) is used to throw material sideways off the belt.

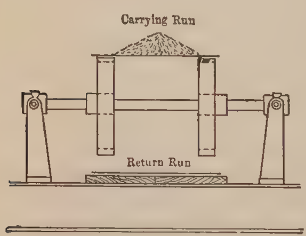


Fig. 61.—Sandvik Flat Steel Belt Conveyor.

All fabric belts, especially rubber belts, have been very costly in Europe since the Great War, and these steel belts are among the substitutes used there, especially in Germany. They are light, run quietly and when run on rollers are said to take less power than fabric belts. The power required to slide the band in wooden troughs is said to be moderate. At a German chemical works a horizontal conveyor 115 feet long carrying 30 tons of potash salt per hour, a steel band sliding in a wood trough at 200 feet per minute required $1\frac{1}{2}$ horse-power. The return belt slid on wood planks; these and the trough bottom were lubricated with flake graphite. At

another works a conveyor 108 feet long inclined 2°, discharges to another 120 feet long inclined 12°, capacity 40 tons potash salt per hour at 200 feet per minute belt speed. The upper run slides in a wood trough, not lubricated; the empty run returns over rollers. Five horse-power drives the two conveyors, including losses in power transmission and the friction at a scraper plow used for discharging the salt.

The Sandvik belts are used in Germany chiefly for conveying materials which will form a load of considerable depth on the belt and not scatter sideways and spill over the edge. With such substances as raw sugar and potash salts, the carrying capacity of a 16-inch belt may be as much as 800 cubic feet per hour per 100 feet per minute belt speed; this is about double the ordinary rating of a flat belt with free-flowing material.

When the upper run travels on rollers, its capacity in free-flowing material is that of a flat belt. (See Fig. 138.) To get a conveyor capacity beyond that of a 16-inch width, which is the present limit of manufacture, some conveyors have been built in Europe with three parallel strips run with their edges overlapping 3 or 4 inches, but not fastened together. The outer strips are driven by pulleys keyed to the head shaft; the middle strip lying over the others engages a narrower pulley on the head shaft driven by friction from the other two. On the return run the three strips are carried separately, the middle one on its own rollers at a level lower than the outer strips. Recent experience with this three-piece construction has shown it to be troublesome in operation, and it is not likely to be widely used.

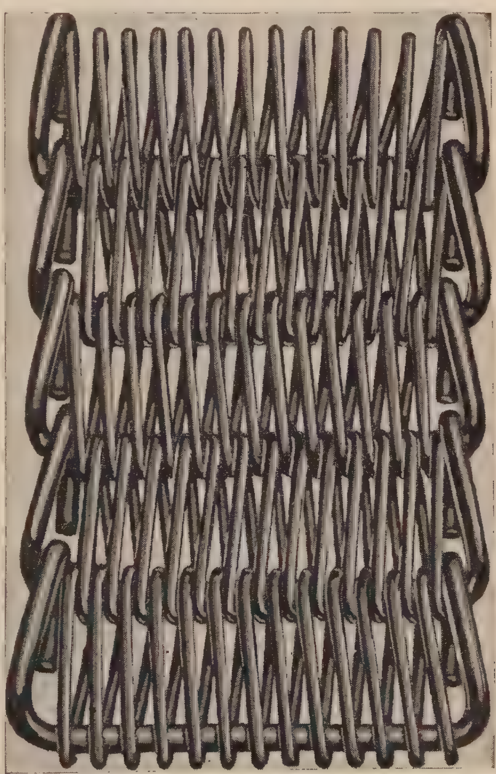


Fig. 62.—Steel Wire Belt used in Europe as a Substitute for Fabric Belt.

Joining the Ends.—To form a splice in the Sandvik band the ends are cut off square, or to form a V (fish-tail splice), then lapped about $2\frac{1}{2}$ inches and secured by two rows of $\frac{3}{16}$-inch rivets $1\frac{1}{4}$-inch pitch.

Steel Mesh Belts have been used in Europe for sixty years or more, but very few have appeared in this country. They consist of coils of wire, round or square, in cross-section, interlaced together and then flattened.

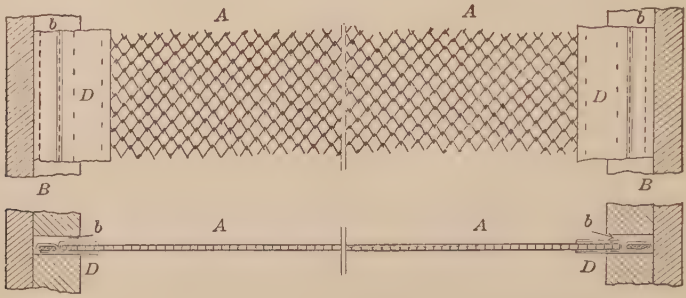


Fig. 63.—Steel Belt of Woven Wire with Fabric Belt Selvages.

As generally used abroad, the coils are joined over wire pins which may be bent at their ends and interlocked to form a selvage or border (Fig. 62) or the pins may be connected by flat links forming a chain selvage. When the coils are interlaced into each other with no joint pins, the pitch shortens

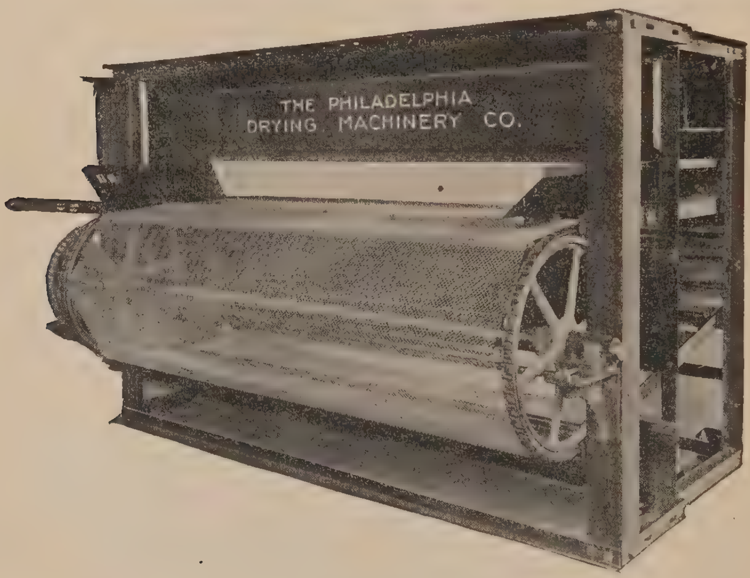


Fig. 64.—Woven Wire Belt with Chain Selvages.

slightly when the coils bend on each other and the belt is more apt to kink and is harder to handle. For that reason a wire strand or a wire rope is sometimes used as a selvage for the wide wire mesh aprons used in drying

machines in this country, or the apron may be held straight and in shape and its edges protected from wear from the tracks on which it slides by the use of an edging of fabric belting as in the Proctor patent of 1900 (Fig. 63) or by chains along each edge as in Fig. 64.

Originally wire mesh belts were used in England and Germany to carry and drain wet coal or sugar beets or to handle packaged goods; but in Germany since the war all fabric belts are so scarce and high-priced that these wire belts are used to carry all sorts of bulk materials, even fine stuff. To lessen the leakage through such belts thin strips of wood or metal are inserted in the flattened coils, or they are covered with thin fabric. These devices must, however, be regarded as makeshifts. They are heavy, and

FIG. 65.—Two Conveyors each with a $\frac{1}{4}$-inch Endless Wire Rope. (Link-Belt Company.)

clumsy, they do not prevent leakage altogether, and there is always spill on the return run of the belt.

In this country wire mesh belts have been used for drying-machine aprons and in a few bakeries to carry and cool freshly baked bread. Several United States patents (for instance, Pattee, 1916) have been issued for belts consisting of combinations of steel tension members with rubber filling, or with layers of fabric. They are not made in a commercial way.

Steel Rope Belts.—A few have been used in the United States. A flexible steel rope spliced endless is laced back and forth between a grooved driving drum and a grooved foot drum with one bend over a sheave which acts as a tightener and also leads the rope from the top of one outside groove on the foot drum to the bottom of the outside groove at the opposite end of the drum. As shown in Fig. 65, it was used to carry and cool freshly baked bread. The necessity for resplicing the rope occasionally was an inconvenience, because few mechanics can do the job right. In later

conveyors built for the same purpose multiple strands of light malleable iron chain have been used instead of the wire rope.

Hemp Rope Belts made of a number of parallel manila ropes stitched and sewed together to a fabric backing have been used abroad for package conveyors. One was used as a baggage conveyor at a railroad station in Paris twenty years ago. The construction has no particular merit either as to low cost or durability in service.

CHAPTER IV

SUPPORTING AND GUIDING THE BELT

Commercial Troughing Idlers.—The development of troughing idlers has been described in Chapter II and reference has been made to a number of styles which embodied various ideas as to how an idler ought to be made, but which have not survived. Those which are on the market to-day may be considered as having passed the test of use. There are, however, styles in machinery just as there are styles in dress, and what happens to be in style this year may not be advertised at all ten years hence. Changes are not always improvements; idlers of old form and simple design are not necessarily inferior to those covered by patents or of recent origin.

The **original three-pulley two-plane idler** shown in Fig. 29 is made

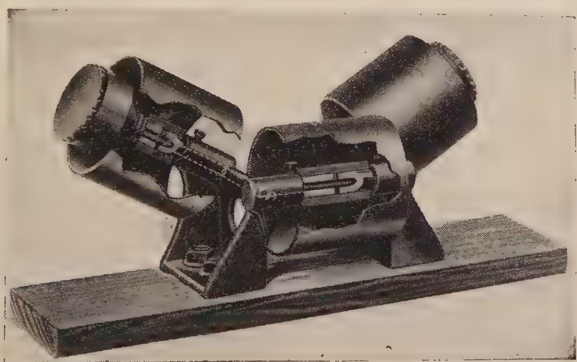


FIG. 66.—3-pulley 2-plane Troughing Idler. (Jeffrey Mfg. Co.)

with improvements, by several manufacturers. It is simple and strong, and there are no gaps between the pulleys where the belt may sag and be pinched. That is because the corners of the pulleys overlap as regards the run of the belt. As made by the Jeffrey Manufacturing Co. (Fig. 66) the three pulleys are mounted on two stands, in which there are grease channels connecting the inclined shafts with the horizontal shaft. This permits two grease cups to lubricate the three pulleys. In the style shown in Fig. 28 (Webster Manufacturing Co., Stephens Adamson Manufacturing Co. and others) the inclined pulleys run on grease-lubricated hollow shafts carried on brackets that are adjustable on the angle bars. The horizontal

pulleys are tight on their shaft; the shaft runs in oscillating bearings, closed at one end and fitted with a grease cup. This type of idler is used on many modern grain conveyors.

Idler Pulleys Tight on the Shaft.—In idlers of the type shown in Fig. 28 the bearings for the horizontal shaft must have some freedom to swivel horizontally and vertically, so that the shaft will be free to turn, even though the stands should be improperly set, or in case the conveyor frame should settle or get out of line. This comment applies to all return idlers and to troughing idlers in which the pulleys are fixed to horizontal shafts.

The Main Belting Co.'s three-pulley two-plane idler is shown in Fig. 67. The horizontal pulley has twice as much contact with the belt as each inclined pulley, and the inclination of the latter is adjustable to three positions—10°, 15°, 20° (Zieber patent, 1916). A wide center pulley has several advantages: 1st—The belt will run straighter; 2d—Since the

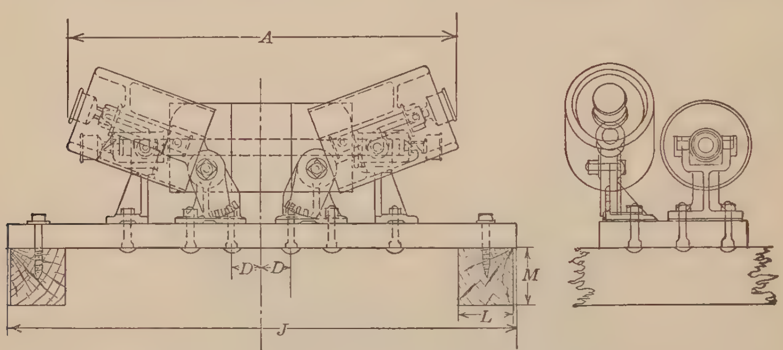


FIG. 67.—3-pulley 2-plane Troughing Idler with Broad Horizontal Pulley. (Main Belting Co.)

horizontal pulley is tight on the shaft, most of the weight of belt and load is carried to closed-end babbitted bearings, and not to the bores of pulleys running loose on a shaft. The adjustment of the inclined pulleys permits the belt to be troughed only as much as is necessary to prevent spill at the loading point and keep material on the belt. Many belts carry loads which do not shift on the belt after they are up to belt speed. In such cases the belt can be troughed as much as necessary to prevent spill or scatter at the loading point, and then run with less troughing for the rest of the distance. In general, the flatter the troughing, the better for the belt; it will run straighter and last longer.

This type of idler is well suited to stitched canvas belts and balata belts; they are stiffer than rubber belts and do not trough so well.

Two Planes or Single Plane.—For the best guiding action, the belt in leading on to a two-plane idler should run as shown by the arrow in Fig. 29; it then gets the guiding and centering action of the horizontal pulley before it touches the inclined pulleys. This is of more importance in narrow belts

than in wide belts, and the precaution is often neglected without particular harm. So far as concerns the life of the belt or the power required to run the conveyor, there is probably no difference between having the pulleys in one plane or two planes.

Single Plane Three-pulley Idlers were first made by Robins (see Fig. 34) and are now made by nearly all concerns in the belt conveyor business. They are listed by most manufacturers for sizes 30 inches and less. Patterns for belts up to 48 inches are in existence, but to a great extent they have been superseded by five-pulley idlers for belts wider than 30 inches. In the original form, the three-pulley idler had two grease cups, in some modern designs, the lubrication of the center pulley is made more certain by using a separate cup for it (Fig. 68) mounted on one of the brackets.

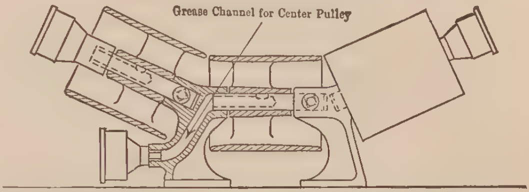


Fig. 68.—3-pulley Troughing Idler with Independent Lubrication for Center Pulley (Link-Belt Company.)

In the Weller Manufacturing Co.'s three-pulley idler (Howard patent, 1920) (Fig. 69) most of the weight of belt and load is carried on a broad center pulley or pulleys fixed to a shaft which runs in babbitted trunnion bearings, each with its own grease cup. The merit of this construction is that the running friction is on babbitted journals and not on a comparatively short hub of a cast-iron pulley with lubrication that is sometimes uncertain. The inclined pulleys act on about one-fourth of the width of the belt at each side and are mounted so that their rims at the bending corner come below the rim of the horizontal pulley; hence the belt can not be pinched there.

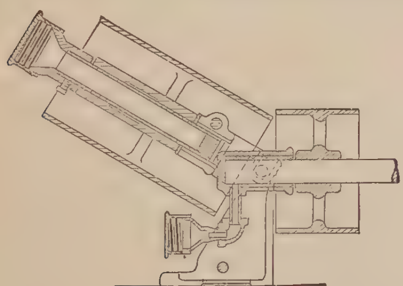


Fig. 69.—3-pulley Troughing Idler with Center Pulley Tight on Shaft and Corner of Inclined Pulley Depressed. (Weller Mfg. Co.)

Idlers for Picking and Sorting Belts.—Three-pulley idlers (Fig. 70) for this service (see p. 193) are made by several manufacturers. The horizontal pulley is made wide so that the material can be carried in a thin layer and thus expose all the pieces to the inspection of the pickers. Spool idlers or flat rolls with occasional concentration can also be used for this work.

Fig. 70.—Troughing Idler with Wide Horizontal Pulley for Picking Belt.

Two-pulley and four-pulley idlers (see p. 14) are listed by some makers. As compared with other idlers, they save in first cost, but are apt to cost more for belt renewals. Since they have no horizontal pulley, the belt is more likely to run crooked (see p. 77), and with the gap between the pulleys under the deepest part of the load on the belt, the belt is always in danger of being squeezed by the pulley rims and split lengthwise.

Five-pulley single-plane idlers are made by nearly all concerns in the business. Current patents refer to methods of making and assembling the cast-iron supporting stands. Robins lists these idlers for all belts from 12 to 60 inches, but other makers do not furnish five-pulley idlers for sizes under 24 or 30 inches.

Five-pulley idlers are generally made with the outside pulleys set at 30° and the intermediates at 15°; the construction is simple and the points of lubrication can be reduced to two cups at the extreme ends of the outer shafts (see Fig. 39) although alternative styles are made by Webster with four cups (see Fig. 71) and by Link-Belt with five cups (see Fig. 75). Web-

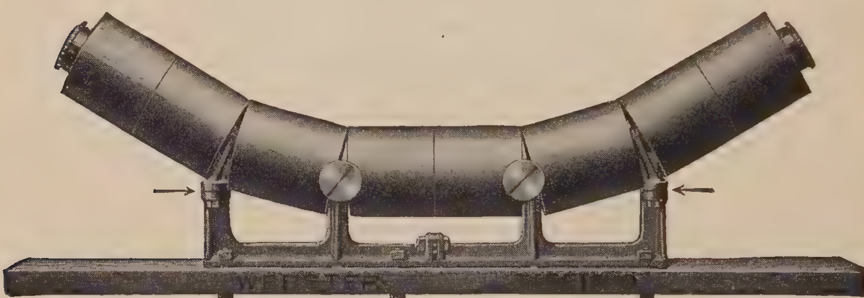


FIG. 71.—5-pulley Idler with Outer Pulleys Adjustable for " Training " the Belt.

ster also makes a five-pulley idler in which the outer and intermediate pulleys on each side can be set slightly out of line with the horizontal pulley (Fig. 71) so as to " train " the belt (see p. 80).

The idea back of the five-pulley idler is that it conforms to the natural curve of the belt without decided bends and is therefore not likely to crack or crease the belt, especially if the belt is not homogeneous in cross-section. (See Fig. 33.) To get that result and avoid gaps between the pulleys, the pulley hubs must be made short so as to leave room for the tops of the cast-iron stands which take the shafts. In most cases, the length of the hub is less than half the pulley face; as a result, the pulleys are more apt to wear loose, run eccentric and rattle than pulleys on three-pulley idlers which have longer hubs. Lubrication of the five separate pulleys from two cups is less certain than in three-pulley idlers where there are only three side outlet holes. So far as carrying capacity of the belt is concerned, there is practically no difference whether the belt runs over three pulleys troughed at 25° or 30° or five pulleys set at 15° and 30° (see Fig. 137).

As to the "steering effect" of five-pulley idlers, see p. 79.
For comment on natural troughing, see p. 76.
For comparison of 15° bends in the belt, see p. 82.

Return Idlers are always flat-faced pulleys mounted on a straight shaft. As made by Robins, the pulleys for belts 36 inches and less in width are spaced by collars along a fixed hollow shaft through which grease is forced by screw cups on the ends of the shaft. In larger sizes, the pulley is a sheet steel tube tight on a $1\frac{3}{16}$-inch shaft which turns in grease-lubricated bearings. Other makers furnish return idlers in all sizes with the pulleys tight on the shaft and the shaft turning in babbitted bearings, which in most cases are of the swivel or trunnion type. It is always advisable to have such bearings self-adjusting so that the shaft will turn freely even if the hangers which support the bearings should be set wrong or get out of alignment. The Stephens Adamson Manufacturing Co. makes return idlers with sheet steel pulleys constructed with recessed heads in which ball bearings are mounted as shown in Fig. 91.

Side-guide Idlers.—Thirty years ago grooved pulleys were used to keep the belt in place on "dish-pan" idlers and on deeply curved spool idlers. They destroyed the edges of belts rapidly. Flat-faced pulleys were common use up to ten years ago; now the style shown in Fig. 72 is more generally used. The ends of the pulley are rounded to avoid cutting the belt if it should ride up on the face of the pulley or get under it as sometimes happens.

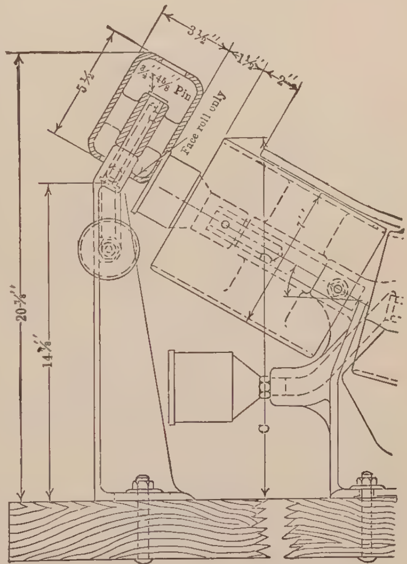


Fig. 72.—Side Guide Pulley with Rounded Edges.

Side-guide pulleys are never required on the return run of the belt unless the carrying idlers are not square with the run of the belt or unless the conveyor is out of alignment with its supports.

Spacing of Belt Idlers.—The spacing of troughing idlers depends upon the weight of belt and load; the heavier the load the closer the supports to prevent the belt from sagging too much. Excessive sag causes internal wear in the belt from stretch of the friction rubber, and wear on its face from the slip of material as the belt passes over each idler. It adds also to the power required to drive the conveyor.

Usual Rules.—Referring to Table 11 and Fig. 73, starting at the head end, the first troughing idler should be from 3 to 5 feet from the center of the head pulley; unless the belt carries an extra heavy load, the material

TABLE 11.—SPACING OF BELT CONVEYOR IDLERS
(Jeffrey Mfg. Co.)

Width of Belt, Inches	Dimension *A* for Material			Dimension *E*	Return Idler Spacing
	Not Over 100 lbs. per Cubic Foot	Over 100 lbs. per Cubic Foot	Sorting Belts Only		
14 to 16	5 ft. 0 in.	4 ft. 6 in.	....	30 ft.	10 ft.
18 to 20	4 ft. 6 in.	4 ft. 0 in.	....	45 ft.	10 ft.
24 to 30	4 ft. 0 in.	3 ft. 6 in.	3 ft. 6 in.	45 ft.	10 ft.
36 to 42	3 ft. 6 in.	3 ft. 0 in.	3 ft. 0 in.	45 ft.	10 ft.
48	3 ft. 6 in.	3 ft. 0 in.	2 ft. 6 in.	......	10 ft.

will not spill as the belt flattens out on the pulley. If the idler is placed too close, the stress in the edge of the belt is too great. Distance *A* for the support of the loaded belt is based upon the practice of several concerns in the belt-conveyor business.

At the loading chute it is important to place the idlers closer together

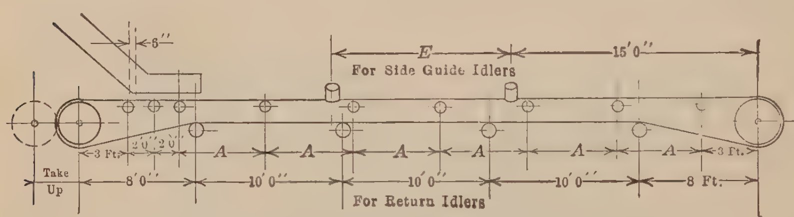


FIG. 73.—Spacing of Belt Idlers. (Jeffrey Mfg. Co.)

to keep the belt well troughed and to prevent the impact of material from deflecting the belt. If the belt sags too much at the loading point material gets under the edges of the chute and skirt-board. Twenty-four inches is a usual spacing at the chute. The heel of the chute should be 4 or 6 inches ahead of an idler so that no lumps can strike directly over the idler. When the belt is free to yield a little under impact, it is not so easily cut by sharp angular lumps of material.

The distance from the last idler to the foot wheel should be at least 3 feet, and a little more is better for the belt. It should be made *A* if possible.

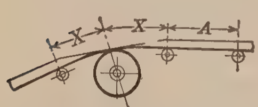


FIG. 74.—Spacing of Troughing Idlers at Hump Pulley.

The distance *X* between a hump pulley and the nearest troughing idler (Fig. 74) should not be less than 3 feet nor more than the regular spacing *A* (Fig. 73).

Return idlers are generally spaced 8 or 10 feet apart. It is not proper to use them as bend pulleys (see p. 73) and they should be set with reference to any floor or supporting frame so that spill from the underside of the belt will not collect there and foul the pulleys of the idler (see p. 177).

Side-guide idlers are a necessary evil with narrow belts on standard troughing idlers (see p. 79); for wide belts which have a good guiding contact with horizontal pulleys, they are not always needed. In any case, use as few of them as possible. Dimension E, (Table 11) is a "standard" spacing for them, the first pair being placed about 15 feet from the head end. They should be set close to a troughing idler and just ahead of it as regards the travel of the belt. If placed midway between idlers, the pressure against the edge of the belt is more likely to turn or fold it over and thus crack the belt.

Idler Spacing for Canvas Belts.—Table 12 gives spacing of idlers recommended for canvas belts.

TABLE 12.—PULLEY SIZES AND IDLER SPACING FOR STITCHED CANVAS BELTS

(Main Belting Co.)

Belt Width, Inches	Usual Plies	Diameter Head Pulley, Inches	Diameter Foot Pulley, Inches	Spacing of Idlers			Spacing of Return Idlers
				Material 50 lbs., Cu. Ft.	Material 75 lbs., Cu. Ft.	Material 100 lbs., Cu. Ft.	
12	4	16–18	12	5 ft. 0 in.	5 ft. 0 in.	5 ft. 0 in.	12 ft. 0 in.
14	4– 5	18–20	12–14	5 ft. 0 in.	5 ft. 0 in.	5 ft. 0 in.	12 ft. 0 in.
16	4– 5	18–20	12–14	5 ft. 0 in.	5 ft. 0 in.	5 ft. 0 in.	12 ft. 0 in.
18	4– 5	18–24	12–18	5 ft. 0 in.	4 ft. 9 in.	4 ft. 6 in.	12 ft. 0 in.
20	4– 5	24–30	14–20	4 ft. 9 in.	4 ft. 6 in.	4 ft. 6 in.	12 ft. 0 in.
22	5– 6	24–30	16–20	4 ft. 6 in.	4 ft. 6 in.	4 ft. 3 in.	12 ft. 0 in.
24	5– 6	24–30	16–24	4 ft. 6 in.	4 ft. 6 in.	4 ft. 3 in.	10 ft. 0 in.
30	5– 8	30–42	18–30	4 ft. 0 in.	4 ft. 0 in.	3 ft. 6 in.	10 ft. 0 in.
36	6– 8	36–42	20–36	3 ft. 9 in.	3 ft. 9 in.	3 ft. 6 in.	10 ft. 0 in.
42	6– 8	36–54	24–42	3 ft. 6 in.	3 ft. 6 in.	3 ft. 3 in.	8 ft. 0 in.
48	6–10	48–60	30–42	3 ft. 6 in.	3 ft. 3 in.	3 ft. 0 in.	8 ft. 0 in.

Elevations of Pulley Rims with Respect to Troughing Idlers.—If the rim of a head pulley is set tangent to the line of the tops of the horizontal pulleys in the idlers there is considerable stretch at the edges of the belt if the troughing is deep. Severe stretch at the edges tends to break down the bond between the rubber cover and the fabric, and is likely to cause a separation of the plies. To reduce the amount of stretch, it is advisable to let the bottom of the belt lift slightly in running onto the head pulley. One rule is to set the rim of the pulley to come half way in the depth of the trough; that is, if the idlers are such as to trough the belt 4 inches deep, the rim of the pulley should be about 2 inches above the top of the horizontal pulleys of the idlers, as seen in the cross-section of the conveyor.

For the same reason, the rim of a hump pulley should be set an inch or two above the lines tangent to the tops of the horizontal pulleys of the idlers on each side of the hump.

Since the belt tension at the foot of a conveyor is generally low, it is

not necessary to set the top of the foot pulley above the line of the horizontal pulleys of the idlers. In fact, it is better to keep the rim of the foot pulley even with the horizontal pulleys to avoid the chance that the belt might lift off the idlers near the foot and be damaged or cut by rubbing against the loading chute or the skirt-boards.

Effect of Increased Spacing.—If the sag in feet of a loaded belt is denoted by H, the span in feet by S, the weight of the loaded belt per foot by w and the pounds tension in the belt by T, then

$$H = \frac{S^2 w}{8T},^{1}$$

or if H and w are constant, then S varies as $\sqrt{T}$, that is, if the tension in the loaded side of a conveyor belt at the head end is four times the tension at the foot end, then for an equal amount of sag between idlers, the idler spacing might be twice as great at the head as at the foot. Considered by itself, this might point the way toward a saving in first cost of the conveyor and some slight economy in power.

There is, however, a more important factor which governs the span between idlers, that is, the tendency of a troughed belt to flatten out in cross-section as shown in Fig. 38. When the trough flattens there is a rearrangement of the load on the belt and a resulting squeeze when the belt is troughed again. The power required for this squeeze may not be much as measured in pounds at each idler, but it causes the friction of the material on itself, and when it is repeated one hundred or more times per minute at each idler, the waste of power is considerable. For instance, in a conveyor 675-foot centers with idlers spaced 3 feet 6 inches apart there are 192 idlers, and at 385 feet per minute travel, a given cross-section on the belt changes its shape $\frac{385}{3.5} = 110$ times per minute. Since there are 192 places where the trough simultaneously flattens out, the result is that in one minute there are 21,120 squeezes exerted on the belt and on the material on it, to bring it back to troughed form again.

Some of the power spent in troughed belt conveyors goes for this work. Where the spacing is close, the change of shape is not great, but if the span is too great, the effect is to hurt the belt and waste power.

Tests to Show Effect of Increased Spacing.[2]—Tests of the 675-foot conveyor mentioned on page 90 showed that with a uniform spacing of 3 feet 6 inches and a steady load of 9 tons per minute or 540 tons per hour, the amperes varied between 20 and 22 with the voltage at 500. These readings are equivalent to 13.5 h.p. and 14.8 h.p. When *one* idler just ahead of the loading point was removed, making the span at that one place 7 feet instead of 3 feet 6 inches, the amperes on numerous tests varied between 50 and 55, with the voltage at 500, corresponding to 33.5 h.p. and 37.0 h.p. That is, over 20 h.p. was required to re-establish the troughed

[1] See Kent's M. E. Pocketbook.
[2] Communicated to the author by Mr. E. C. Auld.

form of the belt at that *one* point. The deflection or sag under load on this 7-foot span measured $1\frac{1}{2}$ inches while the deflection in the adjacent 3-foot 6 inch span measured $\frac{3}{16}$ inch. To lift 9 tons per minute through $1\frac{1}{2}$ inch height takes less than $\frac{1}{10}$ h.p.; it seems clear then that in this case it took more power to squeeze the coal back into troughed form than to convey it 675 feet.

These tests show that the saving in cost of troughing idlers which might be effected by varying the idler spacing is insignificant as compared with the cost of the power required to squeeze the belt and its load back into trough form when the span becomes too great. Since the tendency to flatten out is greater when the belt is deeply troughed, it is quite evident that for a given spacing of idlers more power is required to carry a certain tonnage on a belt over idlers steeply troughed than over idlers with a smaller angle for the side pulleys. Carrying the reasoning still further, properly supported belts run flat or nearly so would require still less power, because the load cross-section does not change. A flat belt will sag more between idlers than a troughed belt because it lacks the stiffening effect of the upturned sides, but with loads based on one-half the maximum capacity of the belt (see p. 144) the difference is not much.

From the fact that the load cross-section does not change on a flat belt, it would be possible to make the idler spacing vary as $\sqrt{T}$ without the loss of power referred to above; but if the sag of a flat belt becomes too great, there would be some disturbance of the load represented by the pile of material on the belt cracking crosswise as it passed over the idler pulley. With ordinary spans, this would amount to less than the disturbance caused by squeezing the material back into trough form; nevertheless it indicates one limitation of the span of belts run flat or slightly troughed.

The ordinary rules for idler spacing represent current practice with troughed belts on standard three-pulley or five-pulley idlers with grease lubrication. Since it is very desirable to keep the load cross-section as nearly constant as possible, it is not wise to increase the spacing of ordinary idlers, because they are relatively cheap, while the cost of power is relatively high. In equipping a conveyor with more expensive idlers, as, for example, with roller bearings, there is some incentive to use greater spacing; but in such cases the saving of power due to the better idlers will justify the use of a greater number of them in order to maintain a constant load cross-section. In general, the advantages are somewhat uncertain when idlers are spaced further apart than is now customary, on the other hand; the disadvantages are quite certain and measurable.

Supporting Belts at Humps and Bends.—Pulleys are generally used to change the direction of belt travel. They should be double-belt straight-face pulleys, 3 or 4 inches in diameter for each ply of belt (see p. 127). This applies to snub pulleys and to pulleys used to change the direction of the loaded belt from an incline to the horizontal, or on the return run. When the return belt is deflected through a considerable angle in passing

from the head pulley to the straight run, or from the straight run to the foot pulley, it is not safe to use a standard return idler. The shaft and the pulleys are both too light for that work.

Troughing Idlers at Humps.—It is not proper to make a hump bend over a standard troughing idler or even a group of them; they are not strong enough, except for light work or where the angle of bend is small There is another objection to the plan of using troughing idlers at humps—that is, the chance that the belt may be injured by the excessive stretch at its edges if the troughed form is maintained over the hump. For instance, if a belt is troughed 6 inches deep on a curve joining a 20° incline with a horizontal run, the length of the edges will be 2 inches greater than the length of the part which lies down on the horizontal pulleys. To resist such deformation, the belt will tend to rise off the horizontal pulleys and rest only on its edges. For the case stated, the stretch is 2 inches, regardless of the radius of curvature of the hump, but obviously the effect on the

FIG. 75.—Natural Sag of Belt on 5-pulley Troughing Idler.

belt is reduced by using a large radius, so that the stretch will be spread over a greater length of belt on the arc.

When a belt flattens out on a pulley used at a hump, the load cross-section changes and power is expended in pushing the material together when the belt is troughed again. This loss of power is the same as that which comes from spacing idlers too far apart (see p. 72) and it is greater than is generally supposed. For that reason, some conveyors have been equipped with troughing idlers at humps, not of the ordinary pattern, but of heavier construction, and by using a number of them, on a large radius of curvature, the tendency to stretch the edges of the belt is reduced and there is less pressure on each idler.

Since the stretch of the edges of the belt varies directly as the depth of the trough it is clear that hump bends over shallow troughing idlers or flared idlers have less effect on the belt than standard idlers which turn the edges of the belt up at 30°.

Natural Troughing.—The Robins stepped-ply belt (p. 12) and the Ridgway hinge-edge belt (p. 17) represent efforts to make belt contour

adapt itself to the contour of three-pulley idlers. The Plummer patent of 1903 (see p. 16) was the first practical expression of the principle of making the idler conform to the natural curvature of the belt. To do this, Plummer proposed to use multiple pulleys with curved faces and thus avoid any bend in the belt. The idea did not come into commercial use; it was followed by various designs of five-pulley idlers which to a great extent match the natural sag of an empty belt as shown in Fig. 75.

In carrying out this idea it is not possible to make any one size of idler fit the curve of all belts because the flexibility for a given width varies with

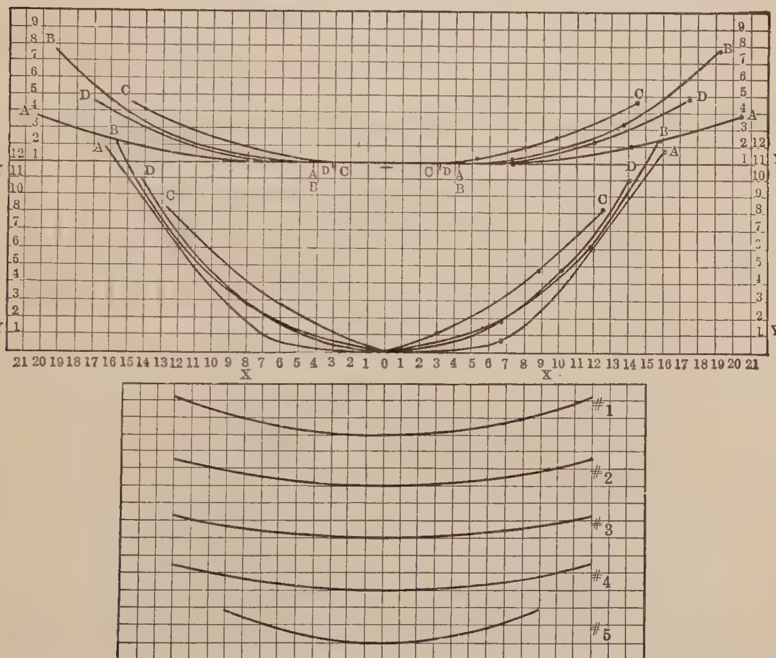


FIG. 76

FIG. 77

FIG. 78

FIGS. 76–77–78.—Natural Sag of Various Rubber Belts. The Horizontal and Vertical Spaces are One Inch.

the weight of duck, number of plies, the thickness of the cover and the grade and the thickness of the layers of friction rubber. All that can be expected is to have the belt lie in a natural curve and avoid sharp bends. In wide belts, the natural curve is generally deeper than the contour of the corresponding width of idler. Fig. 77 shows the natural curvature of four belts when lifted by their edges until the middle merely touched a horizontal surface. In Fig. 76 the same belts were lifted until one-fifth of their width rested on the horizontal surface, this representing the contact of a belt with the middle pulley of a five-pulley idler. In these tests which were

made by the B. F. Goodrich Rubber Co., *A* represents a 42-inch 8-ply belt with $\frac{1}{8}$-inch cover; *B*, a 42-inch 7-ply belt with $\frac{1}{8}$-inch cover; *C*, a 30-inch 7-ply belt with $\frac{1}{8}$-inch cover; *D*, a 36-inch 6-ply belt with $\frac{1}{8}$-inch cover.

In belts narrower than 24 inches the natural sag of an empty belt is generally shallower than the contour of the idler, but when the thickness does not exceed the limits established by good practice (see p. 115) a very slight pressure on the edges of the belt directed toward the center is enough to make its curvature match that of the idler and bring the belt into proper contact with the horizontal pulley of the idler.

Fig. 78 represents the results of tests made by the author.

No. 1 is a standard grade, 24-inch 6-ply belt with $\frac{1}{8}$-inch cover; total thickness, $\frac{7}{16}$ inch.

No. 2 is a belt of higher grade, 24 by 6 by $\frac{1}{8}$ inches; total thickness, $\frac{15}{32}$ inch; it is a little stiffer than No. 1.

No. 3 is a belt of extra quality, 24 inches wide, 5 plies of extra heavy duck, $\frac{1}{8}$-inch top cover and $\frac{1}{32}$-inch pulley cover on the bottom. It has also a layer of "cider-press" cloth to form an anchor for the top cover (see p. 24) and the total thickness is $\frac{9}{16}$ inch. It is less flexible than No. 1 or No. 2.

No. 4 is a high-grade belt used for wet hard work on tailings stackers on gold dredges (see p. 19). It is 24 by 6 by $\frac{1}{8}$ inch with $\frac{1}{32}$-inch bottom-cover; total thickness, $\frac{1}{2}$ inch.

No. 5 is a high-grade trade-marked belt used on grain conveyors, width, 18-inch 4-ply; $\frac{1}{40}$-inch thick rubber covers.

In these five tests the belts were lifted by their edges until the middle merely touched a horizontal surface.

While the five-pulley idler represents in its contour a close approximation to the natural curve of a belt suspended by its edges, it must not be thought that the belts shown in Figs. 76, 77 and 78 will not match the contour of three-pulley idlers or idlers of other form. As has been said above, it needs only a slight pressure, as from inclined troughing pulleys, to make any of the belts assume a trough section much deeper than those shown. Any of these belts will run well with three-pulley idlers troughed at 30° or less.

So far as the action on a good straight-ply belt is concerned, there is little or no difference between "natural troughing" on five-pulley idlers and troughing over three-pulley idlers at 30° or less. There is, however, an advantage in five-pulley idlers for stepped-ply belts and belts in which the width is made up of narrow strips of duck (see p. 15) or where the longitudinal joints between the strips are not properly located to avoid the lines of bending. In belts so made, the smaller angle of bend on five-pulley idlers counteracts, to a degree, the harm done in not having the filler threads in the duck continuous across the lines of flexure. But, on the other hand, the idler lacks something in simplicity, durability, rugged construction, certainty of lubrication and good guiding action. On these points five-pulley idlers may be inferior to three-pulley idlers and idlers of other forms.

Wear of Pulley Hubs and Rims.—In most of the modern three-pulley and five-pulley idlers of the single-plane type an effort is made to get the pulley rims close together to avoid pinching or creasing the belt in the gap between them. (See Figs. 39, 68, 71.) This distance cannot be made too small for fear that the rims of the pulleys will interfere with each other when wear occurs between the bosses of the cast-iron stands and the hubs of the inclined pulleys which rest against them. Interference between rims of pulleys prevents them from turning, and if the pulley rims are cut through by the belt sliding over them, as sometimes happens, the belt may be cut and damaged. It can be prevented by inserting washers between the stands and the pulley hubs before all of the rim clearance has been taken up by wear. The bosses on the stands are not usually finished; although it is customary to finish one end of each pulley hub. If the pulley hubs are central and symmetrical, the rim clearance can be maintained by reversing the pulleys when the hubs are worn on one side.

The pulleys of commercial belt idlers are generally made as light as possible with rims not over $\frac{1}{4}$ or $\frac{5}{16}$ inch thick so that they are lively and revolve easily, and are at the same time cheap. To avoid cracking in the foundry in cooling, and for easy shop work in finishing the rims, the metal used is a soft gray iron. It does not resist wear so well as harder iron, nor is the finished rim so hard as the skin with which the pulley comes from the sand mold. In coke conveyors the abrasive dust wears out the rims of pulleys, and hence some concerns prefer to use troughing and return idlers with no machine finish on the rims of the pulleys but with the foundry skin left on them. Another step in the same direction is the use by some Western mining companies of idler pulleys cast of harder metal in an iron chill. The rim in such pulleys must be quite thick to avoid cracking in cooling. This adds to the cost, and makes the pulley more sluggish in rotation, but under certain operating conditions, where the belt carries wet sharp ore, the longer life of the pulleys justifies the greater expense and perhaps compensates for some added wear on the belt.

Why Belts Run Crooked.—Assuming that the center line of the conveyor is in proper alignment with the end pulleys, a flat belt may run crooked if the axis of the idlers is not square with the run of the belt, or if both edges are not under the same tension, or if the belt was badly made, to begin with. But if the idlers are set square with the travel a normal belt, run flat, tends to run straight. The evidence on this point is that side-guide idlers are never required on the return run of belt conveyors, unless the frame is out of line. However, if the belt is run over troughing idlers of any kind, there comes a tendency to run crooked; this occurred with the spool idlers used on old grain conveyors, and with "dish-pan" idlers also, because if the belt shifted to one side by reason of eccentric loading or bad alignment, it acted as a belt does in running on to a crown-face pulley, that is, moved toward the large part of the pulley.

The action of a belt on troughing idlers is shown in Fig. 79. Considering the right half as illustrating a belt on a spool idler, imagine the belt

composed of strips 1, 2, 3, 4, 5. If the belt is very slack it will take the position 1 2 3 4 5 between idlers, while on the idlers it takes the position *ABCDE*. The outer strip is deflected from 5 to *E* and the line of tension along the center of that section is pulled out of line for a distance *EF*, where *E* indicates the center of the strip on the idler and *F* is directly over the center of the strip at 5. The distance *EF* is a measure of the force parallel to the face of the pulley, which tends to straighten up the pull in the outer strip parallel to the center of the conveyor; hence, section 5, if separate, will move up on the pulley until *E* coincides with *F*. This force can be called *T* tang *P*, where *T* is the tension in strip 5 and *P* is the angle between the plane of the strip on the idler and its plane midway between idlers. That same action in the other separate strips would cause them to shift outward on the pulley face for distances varying with the angle of inclination of the pulley at their place of contact. If the strips were separate, they would run with spaces between them, but considered as parts of a wide belt, they will tend to shift the belt out of center.

Fig. 79.—Steering Effect of Troughing Idlers.

On the left half of Fig. 79, representing a belt on a five-pulley idler, the action is the same, although the revolving surfaces are those of separate cylinders instead of parts of cones.

If the deflecting forces on one side of the belt balance those on the other side, the belt will run straight; if they are unequal, the belt will shift sideways until the balance is restored. Since the force which moves section *F* outward is measured by tang *P* (Fig. 79) it is evident that if the belt between idlers did not sag so much, the tendency to run sideways would be less, because the angle *P* would be less, and if the belt did not sag at all, there would be no tendency to run crooked. This fact explains why the tendency of a belt to run off to one side, can sometimes be corrected by giving the belt more tension on that side by adjusting the take-ups. The diagram explains also why a belt loaded heavily on one side tends to run off on that side. In that case, the belt sags more between idlers and the angle *P* is greater, hence the force outward parallel to the pulley face is greater. We can understand also why belts on 45° troughing idlers are more apt to run crooked than on 30° idlers; with all other conditions the same, the natural elasticity of the belt will cause it to flatten out more between idlers and the angle *P* will be greater the larger the angle of troughing. This elasticity is more pronounced in narrow belts than in wide belts. This explains why it is harder to keep narrow belts straight on any kind of troughing idler.

If a pulley on one side of an idler does not revolve as freely as a pulley

on the opposite side of the center line, the deflecting forces do not balance, and the belt runs crooked.

The sag of belt between idlers depends also upon idler spacing and the weight of the load. In handling heavy material, the belt sags more, and the angle P becomes greater. Hence to keep the belt straight, the troughing idlers must be closer together. There are, of course, other reasons for close spacing for heavy materials.

Steering Effect of Idlers.—The tendency of inclined pulleys to steer a belt or deflect its course, depends upon the angle of inclination and the proportion of belt width in contact with them. It is resisted by the proportion of belt width in contact with the horizontal pulley; hence we can compare various styles of idlers as to their steering effect by multiplying each pulley face by the tangent of the angle it makes with the horizontal, and dividing their sum by the width of the belt. Fig. 80 prepared on this basis shows that there is little difference between the ordinary five-pulley

Angle A	.20 .20 .20 .20 .20 A 1 ½A	.16 .17 .33 .17 .16 A 2 ½A	.33 .33 .33 A 3	.25 .50 .25 A 4
15&30	.338	.275		
10&20	.216	.176		
5			.058	.044
10			.118	.088
15			.178	.134
20			.242	.182
25			.304	.233
30			.384	.289
35			.466	.350
40			.560	.420
45			.666	.500

Fig. 80.—Comparative Steering Effect of Different Troughing Idlers.

idler, column 1, and a three-pulley idler troughed 25° or 30°, column 3, but that when the horizontal pulley has twice the face of the inclined pulley, column 4, troughing at 30° on three pulleys has less steering effect than the standard five-pulley idler with angles 15° and 30°, column 1.

Fig. 80 also shows that the steering effect of a 20° idler with a broad center pulley, column 4, is about half of that of a standard five-pulley idler, column 1. We should therefore expect the belt to run straighter on the former than on the latter. If the angle is reduced to 10°, or even less as in flared idlers, the tendency to steer the belt is very small as compared with five-pulley idlers.

Methods of Making Belts Run Straight.—The old original device to keep belts in place is the side-guide idler bearing against the edge of the belt. With 45° troughing, they were indispensable; on five-pulley idlers and 30° three-pulley idlers narrow belts still need them, but on wider belts where the proportion of belt width acted upon by inclined pulleys is less, they are not so necessary unless operating conditions are bad and unless the methods described below are not effective in keeping the belt straight.

The effect of side-guide idlers is generally bad. They wear the belt at its most vulnerable place and open the way for dirt and wet to get between the plies of fabric. If the belt is badly out of line, the pressure against the idler may be enough to bend or fold the belt for an inch or two all along the edge so that a crack develops there and splits the belt.

Another expedient to keep belts straight is to set the inclined pulleys of the troughing idler with a rake forward in the direction of belt travel. In the Mason patent of 1907 this is done, as shown in Fig. 81; another way with standard three-pulley or five-pulley idlers is to bevel the board which carries the idler so as to tilt it forward (Fig. 82). In either way, the effect is to make the inclined pulleys rub the belt at an angle to the travel and toward its center line. When the belt is centered over the idler these forces balance by equality of belt contact on the pulleys; but if the belt runs off to one side, the skewed pulleys on that side act on more belt surface and tend to push it back again. This method will keep a belt straight where other methods fail, but it is hardly necessary to say that a pulley skewed with reference to the line of belt travel does not revolve as freely as a pulley

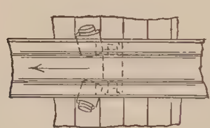


Fig. 81.—Guiding a Belt by Skewing the Inclined Pulleys.

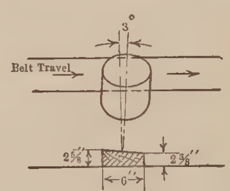


Fig. 82.—Guiding a Belt by Tilting the Idler Stands and Pulleys.

set square with the travel; there must be some waste of power and some wear on the pulley side of the belt, and on the rims of the pulleys.

Before it became customary to bevel the boards on which five-pulley idlers were mounted, it was not uncommon to have serious trouble in getting belts to run straight. An account of twenty-two belt conveyors, nearly all 20 inches wide, installed with five-pulley idlers at a Western smelter in 1914 says: "The belts in the conveying system could not be made to run true on the troughing idlers. To overcome their riding out of position, long boards were fixed at the sides of the belt to guide them and keep them in place." (Bulletin 91 A. I. M. E.) What this did to the edges of the belts is not stated.

In a few makes of three-pulley or five-pulley idlers provision is made to permit the outer pulleys to be set slightly out of alignment with the horizontal pulley, so that by the angular deflection of the pulley, the belt can be steered. (See Fig. 71.)

The Sibley patent of 1916 discloses an attempt to steer the belt automatically. The outer pulleys of a three-pulley idler are mounted on vertical pivots which are connected by cranks and a lever. If the belt is centered with reference to the pivots, the pulleys stand parallel with each

other and square with the belt, but if the belt runs higher up on one pulley, its frictional contact there increases, and the pulley is swung into an angular position which steers the belt back to center again. This device has not had commercial use; dirt, wet, and defective lubrication are some of the conditions which would interfere with its operation.

A number of other schemes have been suggested to steer a belt automatically; they need not be described, but they are mentioned to show the efforts made to overcome a trouble inherent in troughing idlers.

"Training" a Belt.—The stands for troughing idlers are generally provided with slotted holes in the base so that the axis of the pulleys can be adjusted square with the center line of the belt. When a belt persists in running crooked, it is usual to set the stands out of square with the belt so as to force it in place by a rubbing action similar to that described above; although in this case the horizontal pulley of the idler also helps to steer the belt. The process of "training" a belt by adjusting the idlers is, in effect, setting the idler groups out of square one way or the other enough to counteract the tendency of the inclined pulleys to pull the belt crooked.

Besides idler groups set out of square, skewed troughing pulleys, and side-guide idlers, all expedients to make a belt run straight over troughing idlers, there is another which may do even more mischief because its effects are not immediately visible. That is the practice of making it run straight by putting a high tension in it. It is an easy matter for the man in charge of a belt conveyor to correct faults inherent in the idlers or due to poor alignment by screwing the take-ups back (Fig. 113) or loading the suspended pulley (Fig. 112) until the belt at least stays on the idlers. That may mean greater friction losses in the machinery, excessive stretch in the belt, splices pulling apart, frequent resplicing and a short life for the belt. All of these items involve delay, trouble and expense.

In general, the right way to make belts run straight is to use flatter troughing. Just as 30° troughing is better than 45° troughing, so is 20° better than 30° in keeping the belt centered on the idlers and running straight. There is also the collateral advantage of less longitudinal bend in the belt and less tendency to crease or crack it. (See p. 15.) The maximum carrying capacity for shallow troughing is less than for 30° troughing, but the difference as stated in terms of safe loading is not great. (See Chapter VII.)

Internal Stresses in Belts due to Troughing.—All belts suffer internal stresses that tend to shorten their lives when run over troughing idlers, either from the longitudinal bending to the contour of the idler or in the repeated change from the flat position on the end pulleys to or from the troughed position on the idlers. When the troughing angle is steep, rubber belts will stand the bending better than stitched canvas belts, but since the ordinary safe carrying capacity of a belt is not increased much by troughing it beyond 20° (see Figs. 133 and 135) it is to the owner's interest not to use steep troughing with a canvas belt, nor in fact with any kind of belt. In most conveyor installations the price of the belt is a large percentage of

the total first cost of the conveyor, and if the costs of renewals are kept for ten, fifteen, or twenty years it will be found that the charges against the belt for renewals and repairs will be much greater than the sum of the charges against pulleys, idlers, transmission machinery and all the rest of the conveyor apparatus. It pays, therefore, with belts of all kinds, to use large pulleys, well-made idlers and an angle of troughing as flat as will carry an economical load; for there is no doubt that, other things being equal, shallow troughing gives a longer life to belts than steep troughing, and that belts run flat and properly supported last longer than belts troughed at any angle. If shallow troughing or none at all means a wider belt (see p. 86) the extra first cost may be often saved in lower charges for belt renewals and repairs.

Troughing Canvas Belts.—The act of troughing puts a tension on the filler (crosswise) threads, and since these are more likely to be cut and worn by sharp pieces than the warp (lengthwise) threads and since a thread under tension is more subject to that kind of injury than a slack thread, it is evident that a canvas belt run flat will be hurt less by cutting and abrasion than a troughed belt. When a canvas belt is troughed, it is better to keep the inclined pulleys rather far apart so that the places of greatest tension in the filler threads, where the belt bends, will be toward the edges of the belt and not near the center where the abrasion under the loading chute is greatest.

An idler designed especially for canvas belts is shown in Fig. 67. It has a wide horizontal pulley and the line on which the filler threads are flexed is about halfway in from the edge toward the middle of the belt.

Comparison of Troughing Effect, Three-pulley or Five-pulley Idlers.—In one respect a three-pulley idler with inclined pulleys set at 15° stresses the belt less than a standard five-pulley idler which bends the belt 15° at two places on each side of the middle. In the former, the stretch of the filler threads is spread over a length equal to one-half the width of the belt, but on a five-pulley idler the stretch at A (Fig. 133) is confined to the distance BC and the stretch at D comes on the length of thread CE. Both BC and CE are less than one-half the width of the belt; hence for equal elongation of the lower plies at the point of bend the tension in the filler threads may be greater on the five-pulley idler and the tendency to crack greater. In other words, the greater the length of the stressed portion, the less is the stress for a given deformation. Reasoning from this, the tendency to crack the belt on a three-pulley idler is least when the face of the horizontal pulley is about twice the face of each inclined pulley.

Canvas Belts on Idlers Designed for Rubber Belts.—Well-made stitched canvas belts saturated with Class 1 compounds (see p. 47) and used for handling bulk materials will run on five-pulley idlers or 30° three-pulley idlers when the belt is 30 inches or wider, and the plies not over 6; belts between 18 and 24 inches in 4- or 6-ply will not run well on standard idlers. They should not be troughed over 20° and 6-ply belts narrower than 18 inches should not be troughed over 15°; they will not lie down on

standard idlers, but will run crooked unless held in place by side-guide idlers, and these are apt to ruin the edge of the belt.

Spool or Flared Idlers.—The old-time spool idler with its deep concavity had certain defects (see p. 11) and has passed out of existence, but a modified form of it is still in use. In the cement region of Pennsylvania and New Jersey it has survived the competition of three-pulley and five-pulley idlers. As made there by local shops and by several cement companies, it has a cast-iron center pulley with two cast-iron bell-shaped ends which lift the edge of the belt an inch or two. (Fig. 83.) The middle section and the two end sections are tight on the shaft, consequently there is always some wear on the pulley side of the belt near its edges and on the bell-shaped ends, but this wear is often less than might be supposed. In most cases the life of the belt is determined by the more serious wear on the

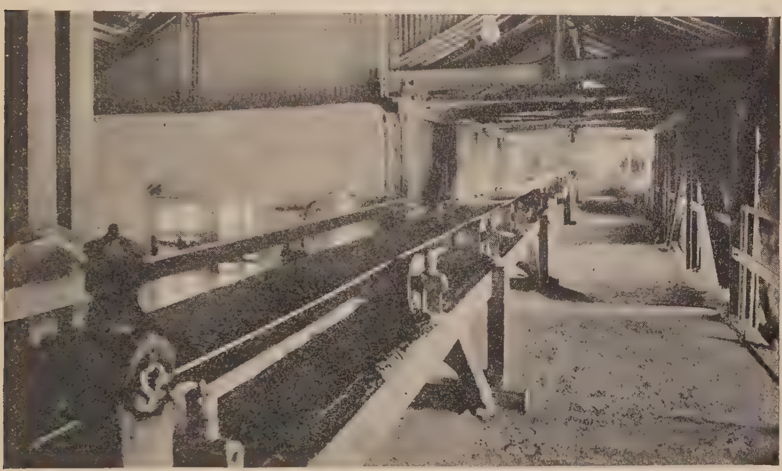


FIG. 83.—24-inch Belt on Cast-iron Spool Idlers Carrying Cement Clinker.

upper surface. A 24-inch belt, 700-feet centers, horizontal, installed in 1908 is still in use (1922) carrying crushed coal with the original flared idlers still in place.

Belts carried by flared idlers are nearly flat, the slight lift of the edges serving to prevent loss of material in transit, but not permitting the belt to convey so much as a belt run over multiple-pulley idlers. On the other hand, the belts run straight without dependence on skewed pulleys or side-guide idlers. Still more important, the lubrication is simple, it is confined to babbitted bearings in which the shaft turns and it is more certain than the greasing of a number of loose pulleys through one, two or three hollow shafts each with its side outlet holes.

Flared idlers used in damp and dirty places may accumulate crusts of material which may injure the belt. For this reason, and because the carrying capacity is less, they are not so general in their application as three-

pulley and five-pulley idlers. In any case, the obvious advantages of the flared idler in simplicity of construction must be weighed against its equally obvious limitations.

The Uniroll idler of the Link-Belt Company is a commercial development of the flared idler described above. It consists of a steel tube fastened to malleable-iron bell-shaped ends and mounted on a straight shaft supported in trunnion bearings. The bearings are made with oil-wells for chain-oiling (Fig. 84) or with roller bearings or with screw cups for grease lubrication. Figure 85 shows the assembly of a carrying idler and of a return idler. Tables 13 and 14 give the principal dimensions.

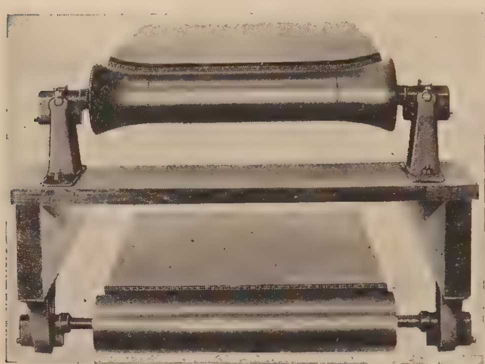


Fig. 84.—Uniroll Flared Idler with Oil Lubrication (Link-Belt Company.)

For carrying capacities of belts on Uniroll idlers, see p. 146.

Troubles with Multiple Pulley Idlers have led some concerns to discard them and substitute plain flat pulleys with no troughing. Figure 86 shows a 24-inch 6-ply belt carrying 50 tons of crushed limestone per hour on flat idlers at 225 feet per minute. The vice-president and superintendent of the company operating it says: " My personal opinion based on experience with conveyor belts is that no type of troughing idler is desirable because lubrication is, as a rule, not properly attended to and the pulleys are allowed to stick, wear flat and then cut the belt. Troughing idlers all have the tendency to break a belt, due to the constant bending between the flat head and tail pulleys and the troughing idlers. At our plant here I have adopted the flat belt and figure it wide enough to have a clear space of 6 inches on each side of the material. With such a width it hardly ever happens that any stones or other material will roll off." " The trouble with belt conveyors, as a rule,

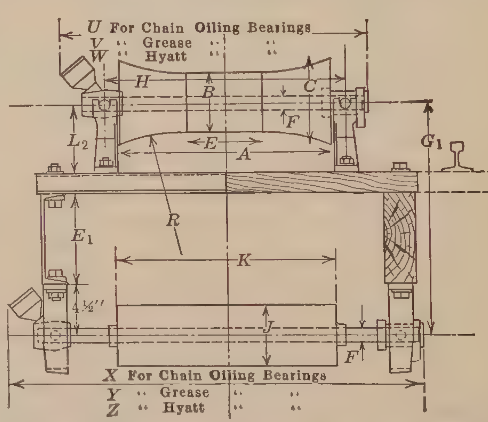


Fig. 85.—Dimensions of Uniroll Idlers and Return Idlers.

TABLE 13.—DIMENSIONS OF UNIROLL IDLERS

(Link-Belt Company)

CARRYING IDLERS

	Width of Belt, Inches										
	12	14	16	18	20	24	30	36	42	48	54
A	$1'\,10\frac{1}{2}''$	$1'\ 2\frac{1}{2}''$	$1'\ 4\frac{1}{2}''$	$1'\ 6\frac{1}{2}''$	$1'\ 8\frac{1}{2}''$	$2'\ 0\frac{1}{2}''$	$2'\ 6\frac{1}{2}''$	$3'\ 0\frac{1}{2}''$	$3'\ 6\frac{1}{2}''$	$4'\ 0\frac{1}{2}''$	$4'\ 6\frac{1}{2}''$
B	$4''$	$4''$	$4''$	$5\frac{1}{4}''$	$5\frac{1}{4}''$	$5\frac{1}{4}''$	$5\frac{1}{4}''$	$5\frac{1}{4}''$	$6\frac{3}{4}''$	$6\frac{3}{4}''$	$6\frac{3}{4}''$
C	$6\frac{1}{8}''$	$6\frac{1}{8}''$	$6\frac{1}{8}''$	$7\frac{3}{8}''$	$7\frac{3}{8}''$	$7\frac{3}{8}''$	$7\frac{3}{8}''$	$7\frac{3}{8}''$	$9\frac{1}{2}''$	$9\frac{1}{2}''$	$9\frac{1}{2}''$
E	$4\frac{1}{2}''$	$6\frac{1}{2}''$	$8\frac{1}{2}''$	$6\frac{3}{4}''$	$8\frac{3}{4}''$	$1'\ 0\frac{3}{4}''$	$1'\ 6\frac{3}{4}''$	$2'\ 0\frac{3}{4}''$	$2'\ 3\frac{3}{4}''$	$2'\ 9\frac{3}{4}''$	$3'\ 3\frac{3}{4}''$
R	$7\frac{1}{8}''$	$7\frac{1}{8}''$	$7\frac{1}{8}''$	$1'\ 3\frac{3}{4}''$	$1'\ 3\frac{3}{4}''$	$1'\ 3\frac{3}{4}''$	$1'\ 3\frac{3}{4}''$	$1'\ 3\frac{3}{4}''$	$1'\ 5\frac{3}{8}''$	$1'\ 5\frac{3}{8}''$	$1'\ 5\frac{3}{8}''$
F	$\frac{15}{16}''$	$\frac{15}{16}''$	$\frac{15}{16}''$	$1\frac{3}{16}''$	$1\frac{3}{16}''$	$1\frac{3}{16}''$	$1\frac{3}{16}''$	$1\frac{3}{16}''$	$1\frac{7}{16}''$	$1\frac{7}{16}''$	$1\frac{7}{16}''$
H	$1'\ 3''$	$1'\ 5''$	$1'\ 7''$	$1'\ 9''$	$1'\ 11''$	$2'\ 3''$	$2'\ 9''$	$3'\ 3''$	$3'\ 10''$	$4'\ 4''$	$4'\ 10''$
L_2	$6''$	$6''$	$6''$	$6''$	$6''$	$6''$	$6''$	$6''$	$7''$	$7''$	$7''$
U	$1'\ 8\frac{3}{8}''$	$1'\ 10\frac{3}{8}''$	$2'\ 0\frac{3}{8}''$	$2'\ 2\frac{1}{4}''$	$2'\ 4\frac{1}{4}''$	$2'\ 8\frac{1}{4}''$	$3'\ 2\frac{1}{4}''$	$3'\ 8\frac{1}{4}''$	$4'\ 3\frac{7}{8}''$	$4'\ 9\frac{7}{8}''$	$5'\ 3\frac{7}{8}''$
V	$1'\ 8\frac{3}{4}''$	$1'\ 10\frac{3}{4}''$	$2'\ 0\frac{3}{4}''$	$2'\ 4\frac{1}{2}''$	$2'\ 6\frac{1}{2}''$	$2'\ 10\frac{1}{2}''$	$3'\ 4\frac{1}{2}''$	$3'\ 10\frac{1}{2}''$			
W	......	......	......	$2'\ 5\frac{1}{2}''$	$2'\ 7\frac{1}{2}''$	$2'\ 11\frac{1}{2}''$	$3'\ 5\frac{1}{2}''$	$3'\ 11\frac{1}{2}''$	$4'\ 7\frac{1}{2}''$	$5'\ 1\frac{1}{2}''$	$5'\ 7\frac{1}{2}''$
M	47	50	52	62	65	71	82	87			
N	63	67	70	82	86	95	109	118			
E_1	$7''$	$7''$	$8''$	$8''$	$8''$	$8''$	$10''$	$10''$	$12''$	$12''$	$12''$

Plank bases for stands $1\frac{3}{4} \times 5\frac{3}{4}''$ up to $42''$, $1\frac{3}{4} \times 7\frac{3}{4}''$ for $42''$ and over.
Channel bases for stands $6''$, 8 lb. up to $42''$, $8''$, $11\frac{1}{4}$ lb. for $42''$ and over.

NOTE.—U is for chain oiling, V is for grease-cup, W is for Hyatt roller bearings.
M is weight (lbs.) of complete idler including plank base.
N is weight (lbs.) of complete idler including channel base.

TABLE 14.—DIMENSIONS OF UNIROLL IDLERS

(Link-Belt Company)

RETURN IDLERS

	Width of Belt, Inches										
	12	14	16	18	20	24	30	36	42	48	54
J	$4''$	$4''$	$4''$	$5\frac{1}{2}''$	$5\frac{1}{4}''$	$5\frac{1}{4}''$	$5\frac{1}{4}''$	$5\frac{1}{4}''$	$5\frac{1}{4}''$	$5\frac{1}{4}''$	$5\frac{1}{4}''$
K	$1'\ 1''$	$1'\ 3''$	$1'\ 5''$	$1'\ 7''$	$1'\ 10''$	$2'\ 2''$	$2'\ 8''$	$3'\ 2''$	$3'\ 8''$	$4'\ 2''$	$4'\ 8''$
F	$\frac{15}{16}''$	$\frac{15}{16}''$	$\frac{15}{16}''$	$1\frac{3}{16}''$	$1\frac{3}{16}''$	$1\frac{3}{16}''$	$1\frac{3}{16}''$	$1\frac{3}{16}''$	$1\frac{7}{16}''$	$1\frac{7}{16}''$	$1\frac{7}{16}''$
X	$2'\ 5\frac{3}{8}''$	$2'\ 7\frac{3}{8}''$	$2'\ 9\frac{3}{8}''$	$2'\ 11\frac{3}{8}''$	$3'\ 1\frac{3}{8}''$	$3'\ 5\frac{3}{8}''$	$3'\ 11\frac{3}{8}''$	$4'\ 5\frac{3}{8}''$	$4'\ 11\frac{3}{8}''$	$5'\ 5\frac{3}{8}''$	$5'\ 11\frac{3}{8}''$
Y	$2'\ 5\frac{3}{4}''$	$2'\ 7\frac{3}{4}''$	$2'\ 9\frac{3}{4}''$	$3'\ 1\frac{1}{2}''$	$3'\ 3\frac{1}{2}''$	$3'\ 7\frac{1}{2}''$	$4'\ 1\frac{1}{2}''$	$4'\ 7\frac{1}{2}''$			
Z	......	......	......	$3'\ 2\frac{1}{2}''$	$3'\ 4\frac{1}{2}''$	$3'\ 8\frac{1}{2}''$	$4'\ 2\frac{1}{2}''$	$4'\ 8\frac{1}{2}''$	$5'\ 3\frac{1}{2}''$	$5'\ 9\frac{1}{2}''$	$6'\ 3\frac{1}{2}''$
G_1	$1'\ 8''$	$1'\ 8''$	$1'\ 9''$	$1'\ 9''$	$1'\ 9''$	$1'\ 9''$	$1'\ 11\frac{1}{8}''$	$1'\ 11\frac{1}{8}''$	$2'\ 1\frac{5}{8}''$	$2'\ 1\frac{5}{8}''$	$2'\ 1\frac{5}{8}''$
S	44	47	49	64	68	73	82	90			

NOTE.—X is for chain oiling, Y is for grease-cup, Z is for Hyatt roller bearings.
S is weight (lbs.) of complete return idler.

is not so much the conveyor itself, but the ignorance of the labor which looks after such machinery around plants like ours. This labor is shiftless, careless and does not pay proper attention to machinery in watching and lubricating it. I would frankly say that few belts are worn out; most of them have to be replaced because of abuse."

For carrying capacity of flat belts, see p. 146.

Deep Troughing or No Troughing.—In 1908 when the five-pulley idler was just coming into use and when most of the three-pulley idlers in use had angles of 30° or 35° an engineer prominent in the belt conveyor business said (Transactions A. S. M. E., Vol. 30): "The only object of troughing is to get an increase of capacity so that when a belt has to be renewed it will cost less money. Unless we get a substantial increase of capacity by a good deep trough, what is the use of troughing at all, thereby sacrificing the simplicity of the single roller that suffices to carry the flat belt." This argument states the case correctly so far as the width of the belt is concerned, because the safe carrying capacity of a belt on three-pulley or five-pulley idlers is double that of a flat belt (see p. 146) and 50 or 60 per cent

Fig. 86.—24-inch Flat Belt Carrying Crushed Stone.

more than when the belt is run over ordinary flared idlers and, consequently, for a given capacity, a conveyor with a narrow troughed belt will cost less than one with a wider flat belt. On the other hand, the flat belt will run straight, will not need side-guide idlers and will not crack lengthwise from troughing. The greater width may permit the use of a wider and better loading chute and a better distribution of the wear over the belt surface, and altogether the wider flat belt may last enough longer than the narrower troughed belt to justify the added expense. The experience of the years since 1908 has shown that belt capacity is not always the most important consideration; simplicity of construction and certainty of lubrication are, in many installations, factors of greater importance.

Lubrication of Idlers.—Quite early in his experience, Mr. Robins discarded oil lubrication (see Fig. 34) because it was difficult with his idler

to prevent belts from being injured by oil dripping or scattering from the pulleys. He found grease more convenient. The pulleys did not turn quite so freely, but the leakage of grease from the pulley bores formed a collar which prevented dust and dirt from getting in. There was usually no drip of grease onto the belt and its use was an advantage in places like stone-crushing plants, where the air was full of dust.

There were, however, some collateral disadvantages which were apparent to designers and users of belt conveyors after some years of experience. In some large plants it requires the full time of several men to keep grease cups filled and the caps screwed down. It sometimes happens that the first evidence of lack of such care is costly damage to the belt. If the attendant misses a cup, no grease is forced through the hollow shaft, a pulley refuses to turn, the belt slides over it, wears the rim thin and finally cuts it away with the result that the belt is cut on the sharp edges. This is more likely to happen on the return run where the grease cups are low and hard to get at and where the idlers are often concealed by the supporting frame or by the protective deck. In this location a failure of the pulleys to turn is not so easily seen as when that trouble occurs on the carrying run.

In some plants equipped with many belt conveyors the cost of grease and the labor of filling and adjusting the cups amounts to a large sum in the course of a year. At one plant that uses about 15,000 feet of wide belts, grease costs over $2000 a year and the labor charge is over $7000 a year (1920). All the belts in this plant do not run every day; if they did, the charges would be considerably more. In another plant that uses about 8000 feet of belt four men are employed in attending to grease cups on the conveyors.

Grease or Oil.—It is well understood that grease lubrication is best for heavy pressures and low velocities and that oil is better suited to lower pressures and higher velocities. This is shown in Fig. 87 based on Tower's experiments (see Kent's M. E. Pocketbook), comparing grease and oil at a temperature of 90° with flooded lubrication. At pressures over 500 pounds per square inch there is not much difference between grease and oil, whether the journal speed is 157 feet or 471 feet per minute, but at lower pressures the difference is quite marked, and below 200 pounds per square inch the coefficients for grease increase more rapidly than those for oil. At 100 pounds per square inch the coefficients were as 2 to 1 within the range of speeds tested.

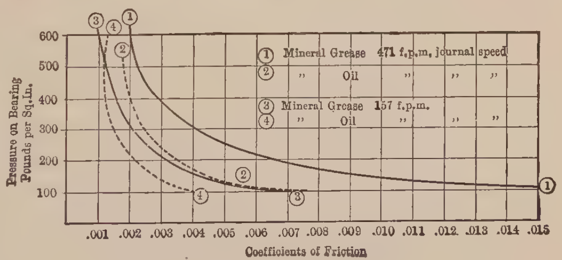


FIG. 87.—Comparison of Coefficients of Friction, Grease or Oil Lubrication.

The Tower experiments were not carried below 100 pounds per square

inch, but since the Galton-Westinghouse experiments (see Kent's M. E. Pocketbook) showed that at velocities below 100 feet per minute and at low pressures the friction varied directly as the pressure, we may say that for copious lubrication at speeds under 157 feet per minute and at pressures below 100 pounds per square inch oil offers one-half the resistance that grease does. In commercial belt idlers the pressure in the bores of the pulleys is much less than 100 pounds per square inch and the speed of the lubricated surfaces is less than 157 feet per minute. Considering these facts and also that a bearing can be "flooded" with oil more easily and with greater certainty than with grease, it seems safe to say that the coefficient of friction for grease lubrication as applied to belt idlers is more than twice the coefficient for oil lubrication—that is, to turn greased idlers requires more than twice as much pull as to turn oiled idlers.

Oil Lubrication for Idlers.—Although the Robins type of single-plane idler was particularly suited to grease lubrication, there have been several designs of idlers of that type with tight pulleys on shafts running in self-oiling bearings, but they never reached the commercial stage. Some of these are shown in Acklin's patent 702273, June 10, 1902, and Bee's patent 800786, October 3, 1905. The bearings for the horizontal shaft are hard to get at; it is difficult to see whether the bearing holds the right quantity of oil, and in the act of filling them oil is apt to get on the pulleys and on the belt. In the self-oiling grain conveyor idler made by several manufacturers no attempt is made to set the three pulleys in line; it follows original grain conveyor practice in having the pulleys mounted in two planes. The pulleys are all tight on their shafts; the horizontal shaft turns in ring-oiling bearings that carry a supply of oil and in which the oil level can be seen. The troughing idler (Bee patent 699477, May 6, 1902) is tight on a stud which runs within a bearing submerged in oil and which can easily be inspected and filled. The mounting is like that of Fig. 27 or Fig. 28.

In flat idlers or idlers of the Uniroll type it is easy to take advantage of oil lubrication. The through shaft which carries the pulleys runs in babbitted bearings (Fig. 84) fitted with oil rings or chains that dip down into an oil well in the base of the casting. This contains enough oil to last some months; the level of the oil and the actual lubrication of the shaft can be seen by turning aside a spring cover on the bearing.

The pull required to move a belt over idlers of this type is not over 60 per cent of that required for five-pulley idlers with grease lubrication.

Ball-bearing Idlers were made in a tentative way for years by several manufacturers; in most of them the cast-iron pulleys were bored and fitted with a self-contained ball bearing at each end of the hub. These required a higher grade of shop work than is usually put on belt idlers. They were expensive and did not sell well.

Idlers with Sheet-steel Pulleys.—In 1912 J. L. Wentz patented a simpler three-pulley idler in which there were no through shafts. Each of the three sheet-steel pulleys (Fig. 88) had a stamped steel head recessed to receive a ball bearing, the inner race of which was carried by a lug rigidly

fastened to the supporting frame of the idler. Not many of these idlers have been made.

The Unit Carrier of the Stephens-Adamson Co. (Aurora, Ill.) represents the development of a ball-bearing idler into commercial form. The "single unit" (Fig. 89) consists of a stamped steel pulley with each end recessed to receive a ball bearing with hardened races; a pressed steel yoke forms a stand for the pulley; a slot at the top edge fits the milled end of the pulley shaft and prevents it from turning. The units can be assembled into groups of 2, 3, 4 or 5 to form idlers of various widths and different angles of troughing (Fig. 90). The construction of the pulley is shown in Fig. 91. The ball bearing is placed in a deep depression in the end of the pulley and a felt packing ring R compressed between two washers helps to keep out dirt and prevent the escape of lubricant with which the bearing is packed. A square shaft holds the felt ring and the washers and also the inner ball-race from rotating.

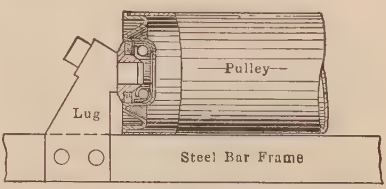


FIG. 88.—Detail of 3-pulley Ball Bearing Idler without Through-shafts. (Robins' Conveying Belt Co.)

FIG. 89.—Ball-bearing Sheet-steel Pulley with Sheet-steel Stand. (Stephens-Adamson Mfg. Co.)

These idlers are light and strong and can be sold at a price to compete with plain grease-cup idlers. The bearings are filled with grease when shipped and if opened and repacked at regular intervals of some weeks or months they run well and last long.

The manufacturers advertise that these idlers save 25 per cent. of power in horizontal conveyors of ordinary length, and since they do not require attention frequently, they save also in attendance and cost of lubricant. When it is necessary to repack the bearings with grease it is more convenient

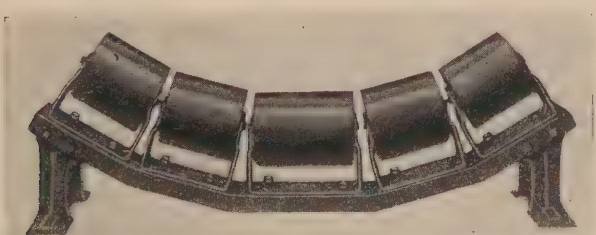


FIG. 90.—5-pulley Troughing Idler Built of Assembled Units. (Stephens-Adamson Mfg. Co.)

in most cases to remove the idler entirely and slip a spare set mounted complete in place of it. This is true especially of wide belts, because it is inconvenient to remove the pulleys and do the repacking under the belt.

In long conveyors the saving of power is more than 25 per cent. A coal conveyor at a mine in western Pennsylvania has a 42-inch belt running at 358 feet per minute on five-pulley ball-bearing idlers spaced 3 feet 6 inches and rising 2 feet in 675-foot centers. The conveyor is driven by tandem pulleys on the return run near the foot and it has a regular capacity of 540 tons per hour. Repeated tests with recording instruments show 20 h.p. or less when the ball bearings are in good condition. The rule stated on page 97 gives 56 h.p.

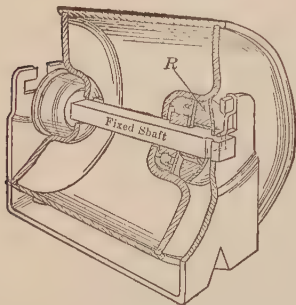


FIG. 91.—Stephens-Adamson Ball-bearing Pulley.

Roller-bearing Idlers.—The hollow spiral roller bearing (Fig. 92) made by the Hyatt Roller Bearing Co., New York, has been applied to belt-conveyor idlers in several ways. In five-pulley idlers the construction used is that shown in Fig. 93. Each pulley is bored to receive the hardened steel sleeve which forms the outer shell of the bearing; steel plates pressed into the bored hole hold the roller bearing in place, and loose steel washers filling the space between the pulley hub and the hub of the stand, take the end-thrust due to the inclined position of the pulley. Four grease cups are used for each troughing idler, so that grease does not have to be forced through the roller bearing in one pulley to reach the next one.

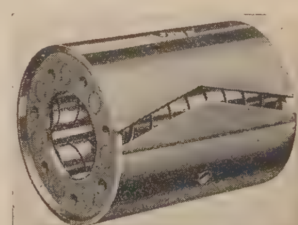


FIG. 92.—Hyatt Roller Bearing.

For return idlers, horizontal carrying idlers of grain conveyors, and Uniroll or flared idlers, the construction is simpler (Fig. 94). The stands carry trunnion bearings closed at one end and bored to receive the shell

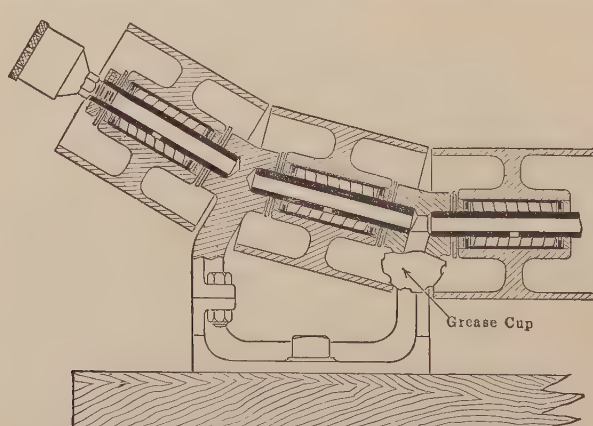


FIG. 93.—Roller Bearings Applied to 5-pulley Troughing Idler.

of the roller bearing. The pulley is tight on the shaft. Each idler then takes two roller bearings instead of five; lubrication is at two points instead

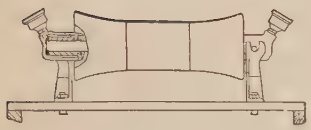


Fig. 94.—Assembly of Flared Idler with Two Roller Bearings.

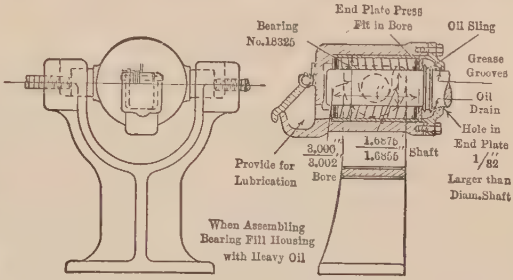


Fig. 95.—Oil Lubricated Roller Bearing for Flared Idler.

of four; none of the grease cups are under the belt and since there is no end-thrust due to inclined pulleys, there is no friction outside of the roller bearings and no wear on stands or on pulley hubs.

Fig. 95 shows a roller bearing for a flared idler designed for oil lubrication and for heavy work. The shaft is $1\frac{11}{16}$ inches, the speed 380 r.p.m. and the load per bearing 400 pounds.

Fig. 96 shows a Hyatt roller bearing applied to a concentrator pulley. Side-guide idlers can be fitted with them in a similar way.

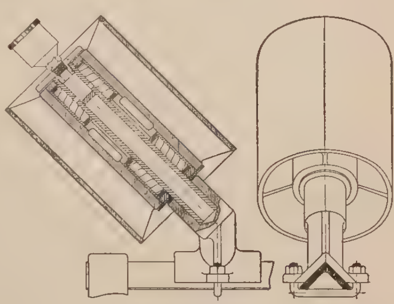


Fig. 96.—Grain Conveyor Concentrator Pulley Fitted with Roller Bearings.

The Stearns idler, made by the Stearns Conveyor Co., Cleveland, Ohio, resembles the Stephens-Adamson unit carrier in being made of 3, 5 or more sheet-steel pulleys carried on independent shafts and mounted on a bent supporting angle. The principal feature of the pulley is a central grease chamber with a spring plunger (Fig. 97). The fixed shaft on which the pulley turns is hollow and has a pipe connection to which a grease gun can be applied. When grease is forced through the shaft the spring plunger is pressed back and the chamber is filled. After that, the spring acts for a period of some weeks or months to force grease into the bearings in the pulley. Instead of the roller bearings shown in Fig. 97, the pulley can be fitted with ball bearings or Babbitt metal bushings.

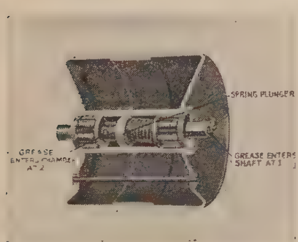


Fig. 97.—Stearns Sheet-steel Pulley with Grease Chamber.

Advantages of Ball-bearing and Roller-bearing Idlers.—As compared with grease-lubricated idlers, idlers with ball or roller bearings have several advantages.

1. Less power to run the conveyor.
2. Less tension on the conveyor belt.
3. Less expense for attendance and for lubricant.
4. Less chance that the idlers will stick tight or drag and injure the belt.

The saving of power cannot be estimated by a comparison of the coefficents of friction for the two kinds of bearing. The reason is that the horse-power formulas in use for grease-lubricated bearings are based on an empirical coefficient of friction (see p. 96) which includes not only the friction in the idler bearings, but also the bending of the belt, getting the material up to speed at the loading point, the slight lifting and squeezing of the load in passing over the idlers, the friction of foot shafts, snub shafts, etc. It is not possible in theory nor in practice to separate the horse-power required for these various items, hence, while we may say that f, the coefficient for journal friction in a greased bearing, is .07 and that a roller bearing will show as low as .0025, it does not follow that the power consumed is in the ratio of those figures or as 1 to 28.

Tests of a 24-inch belt conveyor, 50-foot centers inclined at 15° and heavily loaded to require 10 h.p. showed less than 1 h.p. saving when its idlers were fitted with roller bearings. In these tests about half the power was consumed in motor losses, losses in power transmission, friction of head shaft, two countershafts and foot shaft. In addition, 2 h.p. was required to lift the load; this is independent of the nature of the idlers, so that the saving of power in these tests due to the use of roller-bearing idlers was less than 30 per cent. The saving in power, is of course, more marked in long conveyors where a greater part of the power is used up in overcoming idler friction. The following case illustrates this: A 44-inch belt conveyor 605-foot centers, rising 76 feet, carried stone at 1220 tons per hour. Troughing idlers were like Fig. 93; return idlers also had roller bearings. Tests with recording instruments showed 134 h.p. The manufacturer's calculated rating for this conveyor working with grease-cup idlers was 198 h.p., of which 94 h.p. was for lifting the load through 76-foot vertical height and is therefore to be deducted for any comparison of the qualities of the idlers. The saving of power attributed to the use of Hyatt roller bearings was in this case $104-40=64$ h.p., or 60 per cent of the power due to the friction load, or considering the motor horse-power, the saving was $\frac{64}{198}$ or over 30 per cent.

The following records of recent tests of conveyors fitted with roller bearings in five-pulley idlers will be of interest.

1. 48-inch belt, 387-foot centers, sloped 18°, lift 109 feet, carrying 795 tons run-of-mine coal per hour at 483 feet per minute; 150 h.p. motor used.
   Readings with electrical instruments showed 107 h.p.
   Formula 2 (p. 97) for grease lubrication gives 137 h.p.
   Formula 4 (p. 98) for grease lubrication gives 148 h.p.

2. 48-inch belt, 405-foot centers, sloped 19°, lift 119 feet, carrying 1030 tons run-of-mine coal per hour at 506 feet per minute; 150 h.p. motor used.

Readings with electrical instruments showed 162 h.p.

Formula 2 (p. 97) for grease lubrication gives 190 h.p.

Formula 4 (p. 98) for grease lubrication gives 206 h.p.

3. 42-inch belt, 600-foot centers, sloped 18°, lift 114 feet, carrying 324 tons run-of-mine coal per hour at 242 feet per minute; 100 h.p. motor used.

Readings with electrical instruments showed 52 h.p.

Formula 2 (p. 97) for grease lubrication gives 71 h.p.

Formula 4 (p. 98) for grease lubrication gives 76 h.p.

In another case cited by the makers of these bearings, the reduction in the power required made a saving of $1000 in the price of the motor, and since a belt of lighter ply could safely be used, a reduction of $2700 was made in that item.

Saving in Belt Due to Improved Idlers.—The reduction in the thickness of the belt referred to above may be illustrated by an example. (For explanations of these calculations see pp. 109, 110.) A certain 30-inch 6-ply belt running at 300 feet per minute took 30 h.p. The horse-power pull or $T_1 - T_2$ was therefore $\frac{30 \times 33{,}000}{300} = 3000$ pounds. The head pulley was lagged and with a snub pulley there was a belt wrap of 240°. From Table 21, T_1 was then $3300 \times 1.30 = 4290$-pound pull on the loaded belt. From Table 20, the slack tension T_2 was $\frac{4290}{4.33} = 990$ pounds. The unit tension in the belt was $\frac{4290}{30 \times 6} = 24$ pounds per inch per ply.

If by the use of ball- or roller-bearing idlers the power could be reduced to 25 h.p., then the horse-power pull is 2750 pounds, $T_1 = 3575$ pounds and $T_2 = 825$ pounds; and a 5-ply belt could be used without increasing the unit tension in the fabric because $\frac{3575}{30 \times 5} = 24$ pounds per inch per ply. But if for other reasons it were desirable to use a 6-ply belt, then it would be possible to use its available strength to dispense with the snub pulley and use a belt wrap of 180°. For that condition $T_1 = 2750 \times 1.50 = 4125$ pounds (Table 21) and $T_2 = \frac{4125}{3} = 1375$ pounds (Table 20). That is, by increasing the slack tension by 385 pounds the 6-ply belt will pull the load without the use of a snub pulley and there will be no reverse bend in the belt.

Saving in Repairs and Replacements.—The freedom with which ball-bearing or roller-bearing idlers turn means more than a saving of power; it means that there will be less rubbing on the rims of the pulleys, fewer replacements of pulleys and less chance of cutting and scraping the belt over pulley rims that have been worn thin and cut through. Replace-

ment of idlers is not a serious matter in most plants, but in others, even where gritty material is not handled, grease-cup idlers wear out regularly. In one plant that uses about 8000 feet of belt for conveying run-of-mine and crushed coal, the replacements of idlers for worn-out and broken pulleys and stands per year for a number of years averaged over 15 per cent of the number in use.

Saving in Labor and Attendance.—They save also in attendance and cost less for grease. It is estimated that for one year of ordinary service the following quantities of grease are required for each cup of grease-lubricated idlers: No. 2 cup, 6 pounds; No. 3 cup, 12 pounds; No. 4 cup, 20 pounds. The time needed to fill each cup, over and above the ordinary screwing down, will average about two hours per year per cup.

On the cost of grease and the filling of grease cups see also p. 87.

When roller bearings are fitted to five-pulley idlers (Fig. 93) there is a saving in power which in large conveyors becomes very important (see instances above). There is also a saving in attendance and in the amount of lubricant required, because the bearings need only a small quantity of grease and the cups need not be screwed up often. In this respect alone roller-bearing idlers bring a good return on the investment over and above the saving in power. In plants with only one or two belt conveyors the machine is looked after with care and perhaps some pride as well; then the ordinary grease-cup idlers may receive the attention they deserve. But in a large plant with many conveyors the identity and importance of separate conveyors are lost. To fill and screw up grease cups regularly and faithfully in such a place calls for the steady work of one or more men and the attention which these men give to their work greatly influences the cost of repairs, the cost of shut-downs and the cost of carrying a ton of material. A routine job of that kind is not easy to supervise and consequently it is often shirked.

Roller Bearing Idlers with Oil Lubrication.—A still greater saving of power and maintenance may be expected when the belt is carried on straight or flared idlers and when the roller bearings are lubricated with oil, as shown in Fig. 84. The oil is carried in the base of the bearing and at a low level so that the rollers are not flooded; oil slings prevent leakage along the shaft, there are fewer moving parts and those are of light weight, and all the load is supported by the roller bearings. In multiple-pulley idlers, however, there is always a component of the load represented by end-thrust on pulley hubs which cannot be taken on the roller bearings.

The covered oil well shown in Fig. 84 permits the level of the oil to be seen quickly and when new oil is needed it can be put in easily. It must be said, however, that oil-lubricated bearings are not so simple as those lubricated with grease, and in the type of bearing with a lid shown in Fig. 95 it is not easy to keep the oil clean when the conveyor handles fine material or works in a dusty place.

On the application of improved idlers to long-distance conveying, see p. 107.

Idlers with Tapered Roller Bearings.—There are several advantages in the tapered roller bearings made by the Timken Roller Bearing Company (Canton, Ohio) and others:

1. In the bores of inclined troughing pulleys, the axial thrusts are taken on the hardened roller races and not by the hubs of pulleys or separate washers.

2. The inner races or cones are tight on the shaft or stud and the latter does not turn, hence in idlers with long horizontal shafts, the weight of the rotating parts is less, and there are no uncertainties about alignment as there may be when the pulley is tight on the shaft and the shaft revolves in bearings.

Belts Supported on Runways.—The use of a smooth runway to support a loaded belt (see Fig. 20) is now obsolete, so far as handling bulk materials on fabric belts is concerned; but for carrying dishes, bottles and some other kinds of packaged goods, it is often necessary to use a runway in order to carry the goods smoothly and without danger of upsetting them.

If the loads are heavy or the conveyor long the friction between belt and runway may be too wasteful of power or injurious to the belt. To lessen this friction, it has been suggested (Plummer patent of 1911) that a separate belt, lubricated on the inner side, be used between the carrying run of the conveyor belt and the runway.

CHAPTER V

DRIVING THE BELT

Horse-power to Drive Belt Conveyors.—In general,

$$\text{horse-power} = \text{h.p.} = \frac{\text{pull in pounds} \times \text{speed in feet per minute}}{33{,}000 \text{ foot-pounds per minute}}.$$

In a belt conveyor the output of the motor is expended in overcoming transmission losses up to the conveyor-drive pulley and in putting a certain pull in the belt. The transmission losses will be considered later; the belt pull, which is a measure of the horse-power expended within the conveyor, may be considered as made up of several items: 1. The friction of the idler pulleys in revolving under the weight of belt and material carried; 2. The bending of the belt over the idlers and over the end pulleys; 3. Getting the material up to speed at the loading point; 4. The lifting and slight disturbance of the load in passing over the idlers; 5. The friction of those shafts which are driven through the conveyor belt, such as foot shafts, snub shafts, etc.

Items 2, 3, 4 and 5 are, as a rule, relatively small in amount, and it is difficult to determine them; the usual calculations for horse-power assume a value of item 1 large enough to include 2, 3, 4 and 5 and bring values close to the results obtained from motor readings of conveyors in actual service.

On this assumption the horse-power for a horizontal belt conveyor is

$$\text{h.p.} = \frac{S}{33{,}000}\left(\begin{array}{l}\text{idler friction due to empty belt} \\ +\text{idler friction due to material carried on the belt}\end{array}\right)$$

or

$$\text{h.p.} = \frac{LS}{33{,}000}\left(Xf\frac{d}{D} + Yf\frac{d}{D} + Zf\frac{d}{D}\right), \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where L = length of conveyor in feet, center to center;
S = belt speed in feet per minute;
X = weight of revolving parts of idlers per 1 foot of center distance;
Y = weight of 2 feet of empty belt;
Z = weight of material on 1 foot of belt;
d = diameter of idler bearings;
D = diameter of idler pulleys;
f = coefficient of friction in bearings of idlers.

Experiments with conveyors fitted with grease-lubricated idlers give $f=.35$ and since $\frac{d}{D}$ averages .2, we say $f\frac{d}{D}=.07$.

Since $Z=\frac{\text{Tons per hour} \times 2000}{60 \times S}$, $Zf\frac{d}{D}=\frac{2.33T}{S}$.

To put formula (1) in handy form, X and Y can be expressed as factors for each width of belt. The following shows the method: On a 24-inch belt the pulleys of the troughing idlers of a certain make weigh 40 pounds per set, and at an average spacing of 4 feet along the conveyor they weigh 10 pounds per foot. The return idlers weigh 32 pounds each, or about 3.2 pounds per foot. X is therefore 13.2 pounds and $Xf\frac{d}{D}=13.2\times.07=.93$. A 24-inch 6-ply belt with $\frac{1}{8}$-inch cover weighs about 6 pounds per foot; therefore $Y=12$ and $Yf\frac{d}{D}=.84$ pound. Adding, we get $.93+.84=1.77$ for a factor which represents the idler friction for a 24-inch empty belt; calling this C, we can say

$$\text{h.p.}=\frac{L}{33,000}(CS+2.33T). \qquad (2)$$

This is the formula given by Jeffrey. Table 15 gives, with some interpolations, factors C calculated by Jeffrey for various widths of belt on assumptions of weights of idlers and belts similar to the example given above. These assumptions cover the heaviest belts and closest spacing of idlers likely to be used for the various widths.

If the conveyor is inclined, the horse-power required to lift tons per hour $=T$ through a vertical height $=H$ in feet is

$$\frac{T2000H}{60\times 33,000}=\frac{TH}{990}. \qquad (3)$$

TABLE 15.—DETERMINATION OF HORSE POWER—GREASE-LUBRICATED IDLERS

Width of Belt, In.	C	H.p. for Tripper.	Width of Belt, In.	C	H.p. for Tripper.	Width of Belt, In.	C	H.p. for Tripper.	Width of Belt, In.	C	H.p. for Tripper.
12	.65	1	22	1.85	1.5	32	2.80	3	42	4.15	4.5
14	.75	1	24	2.00	1.5	34	3.15	3	44	4.35	5
16	1.05	1	26	2.15	1.5	36	3.55	3.5	48	4.75	6
18	1.35	1.5	28	2.30	2	38	3.75	3.5	54	5.50	
20	1.70	1.5	30	2.45	2.5	40	3.95	4	60	6.30	

The inclination of the belt adds nothing to the power required to drive it empty, since the downward component of the up-run is balanced by the downward component of the down-run. There is, however, an added stress in the belt when the angle exceeds 2° or 4° (see p. 112 and Table 22).

The horse-power calculated from formulas (2) and (3) does not include

losses in transmission of power from the motor to the drive pulley. To cover these losses and the relatively larger percentage of friction losses in the terminals of short conveyors it is customary to add other factors to determine the total horse-power required to drive the conveyor. The following are the factors as given by Jeffrey: Add 20 per cent for conveyors under 50-foot centers, 10 per cent for 50- to 100-foot centers and 5 per cent for 100- to 150-foot centers. Add for each belt tripper the horse-power given in Table 15. Add 5 per cent for each speed reduction between drive pulley and line shaft, motor or engine, when chain drive, belt drive or cut gears are used; add 10 per cent for each reduction through rough-cast gears.

In the tables published by C. K. Baldwin (Marks, M. E. Handbook, 1st ed., p. 1179) formula (1) has been reduced to the form h.p. $=\frac{KTL}{1000}$ by combining into one term the three terms within the parenthesis. This is done by expressing weights of idlers and weights of belt in terms of belt width and then translating them into terms of belt capacity. For reasonable accuracy, K should be given for materials of various weights, since weight of material determines idler spacing and belt thickness. The line diagrams published by Robins are based on the Baldwin formula but with lines referring to materials of various weights. The formula of the Stephens-Adamson Co. is of the same form but with numerical factors for various weights of material and for ball-bearing and also grease-lubricated idlers, the ratio of the factors for the two kinds being as 3 to 4.

An approximate rule often used for horse-power of belt conveyors is this: 2 per cent of the tons per hour for every 100 feet of length plus 1 per cent of the tons per hour for every 10 feet of vertical lift. This is,

$$\text{h.p.} = \left(\frac{.02L}{100} + \frac{.01\,H}{10}\right)T. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

It is derived from equation (1) by assuming that the total weight of belt and idler parts per foot centers is $\frac{W^2g}{1000}$ where W = belt width in inches and g = weight of material per cubic foot in pounds. Then since

$$T = \frac{3.2W^2gS}{2000 \times 100} \text{ (see capacity formula, p. 144),}$$

we have

$$T = \frac{1.6(X+Y)S}{100} \quad \text{or} \quad X+Y = \frac{T}{.016S};$$

then formula (2) becomes

$$\text{h.p.} = \frac{L}{33{,}000}\left(\frac{.07\,T}{.016} + 2.33\,T\right) = \frac{.02\,TL}{100},$$

to which is added $\frac{.01\,T\,H}{10}$ as an approximation for $\frac{T\,H}{990}$ (see formula 3).

This is a convenient rule and one easy to remember. It gives results that agree closely with those of formula (2) for materials weighing 75 pounds per cubic foot. For materials weighing less than 25 pounds per cubic foot or more than 125 pounds its results are noticeably different, being smaller for light materials and larger for heavy materials as compared with those given by a formula like (2), where the horse-power is made up of the sum of the work required to drive the conveyor empty, plus that required to move the load. It must be said, however, that for light materials formula (2) gives results that are rather high, since the factors in Table 103–1 are calculated for heavier belts and closer spacing of idlers than would be used for materials weighing less than 50 pounds per cubic foot.

Other Formulas for Horse-power.—There are a number of other published formulas for horse-power of belt conveyors, but instead of quoting them with their tables of factors, they will be mentioned briefly as follows:

B. F. Goodrich Rubber Co. Formula (4) (above).

The Goodyear Tire and Rubber Co. A formula similar to (2), but with factors referring to tonnage and belt weight only.

Link-Belt Company. A series of tables based on a formula like (1), but with factors added to cover friction losses at the conveyor terminals.

Main Belting Co. Tables based on a formula similar to (2) giving separate values for empty conveyors and for the loads carried.

Robins Conveying Belt Co. A line diagram which gives results similar to the Baldwin formula (see p. 98).

Stephens-Adamson Manufacturing Co. A comprehensive formula, $\text{h.p.}=\left(\frac{KL+H}{990}\times TJD\right)+P$, and with tables to supply the necessary factors K, J, D and P.

Table 16 gives a comparison of seven rules for the horse-power of belt conveyors as applied to nine specific cases, using in the calculation for each rule the belt capacity as given by the authority for the rule.

Horse-power of Grain Conveyors.—A formula in use for the horse-power of horizontal grain conveyors is $\frac{6}{10}$ horse-power per 1000 bushels per hour per 100 feet of distance. For wheat, which weighs 60 pounds per bushel, this is the same as that given by formula (4)—i.e., h.p. $=2$ per cent of the tons per hour per 100 feet of distance traveled.

Tables Based on Formulas (2) and (3).—Formulas (2) and (3) with Table 15 give all that is necessary to calculate horse-power when L, S, T and H are known. It is, however, convenient at times to have the results of calculations set down in the form of tables, and since formula (2) indicates by separate terms the horse-power for the empty belt and the horse-power for the material conveyed, Table 17 gives values for $\frac{CSL}{33{,}000}$ for the empty belt for conveyors of various widths and lengths for a speed of 100 feet per minute. For other speeds multiply by $\frac{S}{100}$; for other lengths,

TABLE 16.—HORSE-POWER OF BELT CONVEYORS BY SEVERAL RULES

Example of Conveyor	Goodrich		Goodyear		Jeffrey		Link-Belt		Main Belting		Robins		Stephens-Adamson	
	Tons per Hr.	H.p.	Tons per Hr.	H.p.	Tons per Hr.	H.p.	Tons per Hr.	H.p.	Tons per Hr.	H.p.	Tons per Hr.	H.p.	Tons per Hr.	H.p. Note
12″ belt, 200′ c. to c., level, speed 300′ p. m. Coal 50 lbs. cu. ft.	34	1.4	32	1.8	37	1.7	31	3.6	28.5	2.9	34	1.2	30	1.9
16″ belt, 300′ c. to c., level, speed 300′ p. m. Sand 125 lbs. cu. ft.	153	9.2	142	7.4	168	6.4	140	7.8	125	7.6	150	5.0	150	8.0
20″ belt, 400′ c. to c., rises 30′, speed 250′ p. m. Gravel 100 lbs. cu. ft.	160	17.6	148	18.0	175	15.3	160	18.5	150	17.7	160	13.3	150	15.5
24″ belt, 500′ c. to c., rises 60′, speed 400′ p. m. Coal at 50 lbs. cu. ft.	184	29.4	170	31.4	200	31.3	190	32	172	34.4	185	27.9	180	34.4
30″ belt, 400′ c. to c., level, speed 300′ p. m. Stone 125 lbs. cu. ft.	540	43.2	500	30.4	590	25.6	560	28.8	505	30.7	540	21.2	525	32.2
36″ belt, 200′ c. to c., rises 10′, speed 300′ p. m. Lump ore 150 lbs. cu. ft.	933	46.5	864	30.8	1020	31.2	945	32	879	33.2	950	25.8	900	33.6
40″ belt, 223′ c. to c., rises 68′, speed 220′ p. m. Stone 100 lbs. cu. ft.	563	63	502	47.4	616	55.6	562	53.1	528	43.8	562	50.6	528	51.2
48″ belt, 158′ c. to c., rises 11′, speed 140′ p. m. Coke at 30 lbs. cu. ft.	154	6.5	144	5.1	168	6.8	160	9.6	145	8.1	160	6.6	145	......
54″ belt, 734′ c. to c., rises 65′, speed 500′ p. m. R. of M. coal 50 lbs. cu. ft.	1166	246	1200	208	1275	210	1375	271	1095	225	1166	......	1095	......

NOTE.—This table refers to plain grease lubricated idlers.

TABLE 17.—HORSE-POWER BELT CONVEYORS, EMPTY, AT 100 FEET PER MINUTE

(*For Material see Table 18*)

For Other Speeds, S Multiply by $S/100$

Width of Belt—Inches	Length of Conveyor, Feet, Center to Center																									
	10	20	30	40	50	60	70	80	90	100	125	150	175	200	225	250	275	300	325	350	375	400	425	450	475	500
12	.05	.05	.07	.09	.11	.14	.16	.18	.19	.21	.26	.31	.35	.40	.45	.50	.55	.60	.65	.70	.75	.80	.85	.90	.95	1.00
14	.05	.06	.09	.11	.13	.16	.19	.21	.23	.24	.30	.36	.41	.46	.52	.57	.63	.69	.75	.80	.86	.92	.98	1.03	1.09	1.15
16	.05	.09	.12	.15	.18	.22	.26	.29	.31	.34	.42	.51	.57	.64	.72	.80	.88	.96	1.04	1.12	1.20	1.28	1.36	1.44	1.52	1.60
18	.06	.10	.15	.20	.24	.29	.34	.39	.41	.44	.55	.66	.74	.82	.92	1.02	1.12	1.23	1.33	1.43	1.53	1.64	1.74	1.84	1.94	2.05
20	.07	.13	.19	.25	.29	.35	.41	.47	.52	.55	.68	.82	.93	1.04	1.17	1.30	1.43	1.56	1.69	1.82	1.95	2.08	2.21	2.34	2.47	2.60
22	.08	.15	.20	.27	.32	.39	.45	.52	.56	.59	.74	.89	1.00	1.12	1.26	1.40	1.54	1.68	1.82	1.96	2.10	2.24	2.38	2.52	2.66	2.80
24	.09	.16	.21	.29	.34	.42	.48	.55	.60	.64	.80	.96	1.09	1.22	1.37	1.52	1.67	1.83	1.98	2.13	2.28	2.44	2.59	2.74	2.89	3.05
26	.10	.17	.22	.31	.38	.46	.53	.60	.65	.70	.87	1.05	1.18	1.32	1.48	1.65	1.81	1.98	2.14	2.31	2.47	2.64	2.80	2.97	3.13	3.30
28	.10	.18	.25	.33	.40	.48	.56	.64	.69	.74	.91	1.11	1.25	1.40	1.57	1.75	1.92	2.10	2.27	2.45	2.62	2.80	2.97	3.15	3.32	3.50
30	.11	.19	.27	.35	.42	.50	.58	.66	.72	.78	.97	1.17	1.32	1.48	1.66	1.85	2.03	2.22	2.40	2.59	2.77	2.96	3.14	3.33	3.51	3.70
32	.12	.21	.29	.38	.47	.56	.65	.75	.82	.88	1.10	1.32	1.50	1.68	1.89	2.10	2.31	2.52	2.73	2.94	3.15	3.36	3.57	3.78	3.99	4.20
34	.14	.24	.33	.44	.53	.64	.75	.85	.93	1.00	1.25	1.50	1.70	1.90	2.14	2.38	2.52	2.85	3.09	3.33	3.57	3.80	4.04	4.28	4.52	4.75
36	.15	.27	.38	.50	.60	.72	.84	.96	1.05	1.12	1.39	1.68	1.91	2.14	2.41	2.68	2.95	3.21	3.48	3.75	4.02	4.28	4.55	4.82	5.09	5.35
38	.16	.29	.39	.52	.64	.76	.89	1.01	1.11	1.19	1.49	1.79	2.02	2.26	2.54	2.82	3.10	3.39	3.67	3.95	4.23	4.52	4.80	5.08	5.36	5.65
40	.17	.30	.42	.55	.67	.80	.93	1.07	1.17	1.25	1.51	1.88	2.13	2.38	2.68	2.98	3.28	3.57	3.87	4.17	4.47	4.76	5.06	5.36	5.66	5.95
42	.18	.32	.44	.58	.70	.84	.98	1.12	1.23	1.32	1.65	1.98	2.24	2.50	2.81	3.12	3.43	3.75	4.06	4.37	4.68	5.00	5.31	5.62	5.93	6.25
48	.20	.36	.50	.66	.80	.96	1.12	1.28	1.40	1.50	1.87	2.25	2.45	2.86	3.22	3.58	3.94	4.29	4.65	5.01	5.37	5.72	6.08	6.44	6.80	7.15
54	.24	.42	.58	.77	.94	1.12	1.31	1.49	1.64	1.74	2.17	2.61	3.05	3.30	3.71	4.12	4.53	4.95	5.36	5.77	6.18	6.60	7.01	7.42	7.83	8.25
60	.28	.48	.66	.88	1.06	1.28	1.50	1.70	1.86	2.00	2.50	3.00	3.39	3.78	4.25	4.72	5.19	5.67	6.14	6.61	7.08	7.56	8.03	8.50	8.97	9.45

TABLE 18.—HORSE-POWER—HORIZONTAL BELT

(For Empty Conveyor

MATERIAL PER HOUR IN TONS (2000 POUNDS)

Tons per Hour	LENGTH OF CONVEYOR—												
	10	20	30	40	50	60	70	80	90	100	125	150	175
5	.05	.05	.05	.05	.05	.05	.05	.05	.05	.05	.05	.06	.07
10	.05	.05	.05	.05	.05	.05	.06	.07	.07	.08	.10	.12	.13
15	.05	.05	.05	.06	.06	.08	.08	.10	.11	.12	.15	.18	.19
20	.05	.05	.05	.07	.08	.10	.12	.13	.14	.15	.20	.23	.26
25	.05	.05	.06	.08	.10	.12	.14	.16	.17	.19	.15	.29	.32
30	.05	.06	.08	.10	.12	.15	.17	.20	.21	.23	.29	.34	.39
35	.05	.07	.09	.12	.14	.17	.21	.23	.24	.27	.34	.40	.45
40	.05	.08	.10	.14	.16	.19	.23	.26	.28	.30	.38	.45	.52
45	.05	.09	.12	.15	.18	.22	.26	.29	.31	.34	.43	.51	.58
50	.05	.09	.13	.17	.20	.24	.28	.32	.35	.38	.47	.56	.64
55	.06	.10	.14	.18	.22	.27	.31	.36	.39	.42	.52	.62	.70
60	.06	.11	.15	.20	.24	.29	.34	.39	.42	.45	.57	.68	.77
65	.07	.12	.17	.22	.26	.32	.37	.42	.46	.49	.62	.74	.83
70	.07	.13	.18	.23	.28	.34	.40	.45	.49	.53	.66	.79	.90
75	.08	.14	.19	.25	.30	.37	.43	.49	.52	.57	.71	.85	.96
80	.08	.15	.20	.27	.32	.39	.45	.52	.56	.60	.76	.90	1.03
85	.09	.16	.21	.29	.34	.42	.48	.55	.60	.64	.81	.95	1.09
90	.09	.17	.23	.30	.36	.44	.51	.58	.63	.68	.85	1.01	1.16
95	.10	.18	.24	.32	.38	.46	.54	.62	.67	.72	.89	1.06	1.22
100	.10	.18	.25	.33	.40	.48	.56	.64	.70	.75	.94	1.12	1.28
125	.13	.23	.31	.41	.50	.60	.70	.80	.88	.94	1.18	1.40	1.60
150	.15	.27	.38	.50	.60	.72	.84	.96	1.05	1.13	1.41	1.68	1.92
175	.18	.32	.44	.58	.70	.84	.98	1.12	1.23	1.32	1.65	1.96	2.24
200	.20	.36	.50	.66	.80	.96	1.12	1.28	1.40	1.50	1.88	2.24	2.56
225	.23	.41	.56	.75	.90	1.08	1.26	1.44	1.58	1.69	2.12	2.52	2.88
250	.25	.45	.63	.83	1.00	1.20	1.40	1.60	1.75	1.88	2.35	2.80	3.20
275	.28	.50	.69	.91	1.10	1.32	1.54	1.76	1.93	2.06	2 59	3.08	3.54
300	.30	.54	.75	.99	1.20	1.44	1.68	1.92	2.10	2.25	2.82	3.36	3.84
325	.33	.59	.81	1.07	1.30	1.56	1.82	2.08	2.28	2.44	3.06	3.64	4.16
350	.35	.63	.87	1.15	1.40	1.68	1.96	2.24	2.45	2.63	3.29	3.92	4.48
375	.38	.68	.93	1.24	1.50	1.80	2.10	2.40	2.63	2.81	3.53	4.20	4.80
400	.40	.72	1.00	1.32	1.60	1.92	2.24	2.56	2.80	3.00	3.76	4.48	5.12
425	.43	.77	1.06	1.41	1.70	2.04	2.38	2.72	2.98	3.18	4.00	4.76	5.44
450	.45	.81	1.12	1.48	1.80	2.16	2.52	2.88	3.15	3.37	4.23	5.04	5.76
475	.48	.86	1.18	1.56	1.90	2.28	2.66	3.04	3.33	3.56	4.47	5.32	6.08
500	.50	.90	1.25	1.65	2.00	2.40	2.80	3.20	3.50	3.75	4.70	5.60	6.40
600	.60	1.08	1.50	1.98	2.40	2.88	3.36	3.84	4.20	4.50	5.64	6.72	7.68
700	.70	1.26	1.75	2.31	2.80	3.36	3.92	4.48	4.90	5.25	6.58	7.84	8.96
800	.80	1.44	2.00	2.64	3.20	3.84	4.48	5.12	5.60	6.00	7.52	8.96	10.24
900	.90	1.62	2.25	2.97	3.60	4.32	5.04	5.76	6.30	6.75	8.46	10.08	11.52
1000	1.00	1.80	2.50	3.30	4.00	4.80	5.60	6.40	7.00	7.50	9.40	11.20	12.80

CONVEYORS—FOR MATERIAL ONLY

See Table 17)

Feet—Center to Center

200	225	250	275	300	325	350	375	400	425	450	475	500
.07	.08	.09	.10	.11	.12	.13	.14	.15	.16	.16	.17	.18
.14	.16	.18	.20	.22	.24	.25	.27	.29	.31	.32	.34	.36
.21	.24	.27	.30	.33	.36	.38	.40	.43	.46	.48	.51	.54
.28	.32	.36	.40	.43	.47	.50	.54	.58	.61	.65	.68	.72
.36	.40	.45	.49	.54	.59	.63	.68	.72	.76	.81	.85	.90
.43	.48	.54	.59	.65	.70	.75	.81	.86	.92	.97	1.02	1.08
.50	.56	.63	.69	.76	.82	.88	.95	1.00	1.07	1.13	1.19	1.26
.57	.64	.72	.79	.86	.93	1.00	1.08	1.15	1.22	1.29	1.36	1.44
.64	.72	.81	.89	.97	1.05	1.13	1.21	1.29	1.37	1.45	1.53	1.61
.71	.80	.89	.98	1.07	1.16	1.25	1.34	1.43	1.52	1.61	1.70	1.79
.78	.88	.98	1.08	1.18	1.28	1.38	1.48	1.57	1.68	1.77	1.87	1.97
.85	.96	1.07	1.18	1.29	1.40	1.50	1.61	1.72	1.83	1.93	2.04	2.15
.93	1.04	1.16	1.28	1.40	1.52	1.63	1.75	1.86	1.98	1.20	2.21	2.33
1.00	1.12	1.25	1.38	1.50	1.64	1.75	1.88	2.00	2.13	2.26	2.38	2.51
1.07	1.20	1.34	1.47	1.61	1.75	1.88	2.01	2.14	2.28	2.42	2.55	2.69
1.14	1.28	1.43	1.57	1.72	1.86	2.00	2.15	2.29	2.43	2.58	2.72	2.86
1.21	1.36	1.51	1.67	1.83	1.98	2.13	2.29	2.43	2.59	2.74	2.89	3.04
1.28	1.44	1.60	1.77	1.94	2.09	2.25	2.42	2.58	2.74	2.90	3.06	3.22
1.35	1.52	1.69	1.87	2.04	2.21	2.38	2.55	2.72	2.89	3.06	3.23	3.40
1.42	1.60	1.78	1.96	2.14	2.32	2.50	2.68	2.86	3.04	3.22	3.40	3.58
1.78	2.00	2.23	2.45	2.68	2.90	3.13	3.35	3.58	3.80	4.03	4.25	4.48
2.13	2.40	2.67	2.94	3.21	3.48	3.75	4.02	4.29	4.56	4.83	5.10	5.37
2.49	2.80	3.12	3.43	3.75	4.06	4.38	4.69	5.01	5.28	5.64	5.95	6.27
2.84	3.20	3.56	3.92	4.28	4.64	5.00	5.36	5.72	6.08	6.44	6.80	7.16
3.20	3.60	4.01	4.41	4.82	5.22	5.63	6.03	6.44	6.84	7.25	7.65	8.06
3.55	4.00	4.45	4.90	5.35	5.80	6.25	6.70	7.15	7.60	8.05	8.50	8.95
3.91	4.40	4.90	5.39	5.89	6.38	6.88	7.37	7.87	8.36	8.86	9.35	9.85
4.26	4.80	5.34	5.88	6.42	6.96	7.50	8.04	8.58	9.12	9.66	10.20	10.74
4.62	5.20	5.89	6.37	6.96	7.54	8.13	8.71	9.30	9.88	10.47	11.05	11.64
4.97	5.60	6.33	6.86	7.50	8.12	8.76	9.38	10.01	10.64	11.27	11.90	12.53
5.33	6.00	6.78	7.35	8.03	8.70	9.38	10.05	10.73	11.40	12.07	12.75	13.43
5.68	6.40	7.12	7.84	8.56	9.28	10.00	10.72	11.44	12.16	12.88	13.60	14.32
6.04	6.80	7.57	8.33	9.09	9.86	10.63	11.39	12.16	12.92	13.69	14.45	15.22
6.39	7.20	8.01	8.82	9.63	10.44	11.25	12.06	12.87	13.68	14.49	15.30	16.11
6.75	7.60	8.46	9.31	10.16	11.02	11.88	12.73	13.59	14.44	15.30	16.15	17.01
7.10	8.00	8.90	9.80	10.70	11.60	12.50	13.40	14.30	15.20	16.10	17.00	17.90
8.52	9.60	10.68	11.76	12.84	13.92	15.00	16.08	17.16	18.24	19.32	20.40	21.48
9.94	11.20	12.46	13.72	14.98	16.24	17.50	18.76	20.02	21.28	22.54	23.80	25.06
11.36	12.80	14.24	15.68	17.12	18.56	20.00	21.44	22.88	24.32	25.76	27.20	28.64
12.78	14.40	16.02	17.64	19.26	20.88	22.50	24.12	25.74	27.36	28.98	30.60	32.22
14.20	16.00	17.80	19.60	21.40	23.20	25.00	26.80	28.60	30.40	32.20	34.00	35.80

TABLE 19.—HORSE-POWER TO LIFT MATE

(For Horse-power for Level Conveyor

Material per Hour in Tons (2000 lbs.)	Height of Vertical											
	5	10	15	20	25	30	35	40	45	50	55	60
5	.03	.06	.09	.11	.14	.16	.19	.21	.24	.26	.29	.31
10	.06	.11	.16	.21	.26	.31	.36	.41	.46	.51	.56	.61
15	.09	.16	.24	.31	.39	.46	.54	.61	.69	.76	.84	.92
20	.11	.21	.31	.41	.51	.61	.71	.81	.91	1.01	1.11	1.22
25	.14	.26	.39	.51	.64	.76	.89	1.02	1.15	1.27	1.40	1.52
30	.16	.31	.46	.61	.76	.91	1.06	1.22	1.37	1.52	1.67	1.82
35	.19	.36	.54	.71	.89	1.06	1.25	1.42	1.60	1.77	1.95	2.13
40	.21	.41	.61	.81	1.02	1.22	1.42	1.62	1.82	2.02	2.22	2.43
45	.24	.46	.69	.91	1.15	1.37	1.60	1.82	2.05	2.28	2.51	2.73
50	.26	.51	.76	1.01	1.27	1.52	1.77	2.02	2.28	2.53	2.78	3.03
55	.29	.56	.84	1.11	1.40	1.67	1.95	2.22	2.51	2.78	3.06	3.34
60	.31	.61	.92	1.22	1.52	1.82	2.13	2.43	2.73	3.03	3.34	3.64
65	.34	.66	1.00	1.32	1.65	1.97	2.31	2.63	2.96	3.28	3.62	3.94
70	.36	.71	1.07	1.42	1.78	2.13	2.48	2.83	3.19	3.54	3.90	4.25
75	.39	.76	1.15	1.52	1.91	2.28	2.66	3.03	3.42	3.79	4.18	4.55
80	.41	.81	1.22	1.62	2.03	2.43	2.84	3.24	3.64	4.04	4.45	4.85
85	.44	.86	1.30	1.72	2.16	2.58	3.02	3.44	3.87	4.29	4.73	5.15
90	.46	.91	1.37	1.82	2.28	2.73	3.19	3.64	4.10	4.55	5.01	5.46
95	.49	.96	1.45	1.92	2.41	2.88	3.37	3.84	4.33	4.80	5.29	5.70
100	.51	1.01	1.52	2.02	2.53	3.03	3.54	4.04	4.55	5.05	5.56	6.06
125	.64	1.27	1.90	2.53	3.16	3.79	4.43	5.05	5.69	6.32	6.95	7.58
150	.76	1.52	2.28	3.03	3.79	4.55	5.31	6.06	6.82	7.58	8.34	9.09
175	.89	1.78	2.66	3.54	4.42	5.31	6.20	7.07	7.96	8.85	9.73	10.61
200	1.01	2.02	3.03	4.04	5.05	6.06	7.07	8.08	9.09	10.10	11.11	12.12
225	1.14	2.28	3.41	4.55	5.68	6.81	7.96	9.09	10.23	11.37	12.50	13.64
250	1.27	2.53	3.79	5.05	6.32	7.58	8.84	10.10	11.37	12.63	13.89	15.15
275	1.40	2.79	4.17	5.56	6.95	8.34	9.73	11.11	12.51	13.90	15.28	16.67
300	1.52	3.03	4.55	6.06	7.58	9.09	10.61	12.12	13.64	15.15	16.67	18.18
325	1.65	3.29	4.93	6.57	8.21	9.85	11.50	13.13	14.78	16.42	18.06	19.70
350	1.77	3.54	5.30	7.07	8.84	10.61	12.37	14.14	15.91	17.68	19.44	21.21
375	1.90	3.80	5.68	7.58	9.47	11.37	13.26	15.15	17.05	18.95	20.83	22.73
400	2.02	4.04	6.06	8.08	10.10	12.12	14.14	16.16	18.18	20.20	22.22	24.24
425	2.15	4.30	6.44	8.59	10.73	12.88	15.03	17.17	19.32	21.47	23.61	25.76
450	2.28	4.55	6.82	9.09	11.37	13.64	15.92	18.18	20.46	22.73	25.00	27.27
475	2.41	4.80	7.20	9.60	12.00	14.40	16.80	19.19	21.60	24.00	26.39	28.79
500	2.53	5.05	7.58	10.10	12.63	15.15	17.68	20.20	22.73	25.25	27.78	30.30
600	3.03	6.06	9.09	12.12	15.15	18.18	21.21	24.24	27.27	30.30	33.33	36.36
700	3.53	7.07	10.60	14.14	17.67	21.21	24.74	28.28	31.81	35.35	38.88	42.42
800	4.04	8.08	12.12	16.16	20.20	24.24	28.28	32.32	36.36	40.40	44.44	48.48
900	4.55	9.09	13.64	18.18	22.73	27.27	32.82	36.36	40.91	45.45	50.00	54.54
1000	5.05	10.10	15.15	20.20	25.25	30.30	35.35	40.40	45.45	50.50	55.55	60.60

RIALS ON INCLINED BELT CONVEYORS

See Tables 17 and 18)

LIFT IN FEET

65	70	75	80	85	90	95	100	110	120	130	140	150
.34	.36	.39	.41	.44	.46	.49	.51	.56	.61	.66	.71	.76
.66	.71	.76	.81	.86	.91	.96	1.01	1.12	1.22	1.32	1.42	1.52
1.00	1.07	1.15	1.22	1.30	1.37	1.45	1.52	1.68	1.83	1.98	2.13	2.28
1.32	1.42	1.52	1.62	1.72	1.82	1.92	2.02	2.23	2.43	2.63	2.83	3.03
1.65	1.78	1.91	2.03	2.16	2.28	2.41	2.53	2.78	3.04	3.29	3.54	3.79
1.97	2.13	2.28	2.43	2.58	2.73	2.88	3.03	3.34	3.64	3.94	4.25	4.55
2.31	2.48	2.66	2.84	3.02	3.19	3.37	3.54	3.90	4.25	4.60	4.96	5.31
2.63	2.83	3.03	3.24	3.44	3.64	3.84	4.04	4.45	4.85	5.26	5.66	6.06
2.96	3.19	3.42	3.64	3.87	4.10	4.33	4.55	5.01	5.46	5.92	6.37	6.82
3.28	3.54	3.79	4.04	4.29	4.55	4.80	5.05	5.56	6.06	6.57	7.07	7.58
3.62	3.90	4.18	4.45	4.73	5.01	5.29	5.56	6.12	6.67	7.23	7.68	8.34
3.94	4.25	4.55	4.85	5.15	5.46	5.70	6.06	6.67	7.28	7.88	8.49	9.09
4.28	4.60	4.93	5.26	5.59	5.92	6.25	6.57	7.23	7.89	8.54	9.20	9.85
4.60	4.95	5.31	5.66	6.01	6.37	6.72	7.07	7.78	8.49	9.20	9.91	10.61
4.93	5.31	5.69	6.07	6.45	6.83	7.21	7.58	8.34	9.10	9.86	10.62	11.37
5.26	5.66	6.07	6.47	6.88	7.28	7.68	8.08	8.89	9.70	10.51	11.32	12.12
5.59	6.01	6.45	6.88	7.31	7.74	8.17	8.59	9.45	10.31	11.17	12.03	12.88
5.92	6.37	6.83	7.28	7.74	8.19	8.64	9.09	10.00	10.91	11.82	12.73	13.64
6.25	6.27	7.21	7.68	8.17	8.64	9.12	9.60	10.56	11.52	12.48	13.44	14.40
6.57	7.07	7.58	8.08	8.59	9.09	9.60	10.10	11.11	12.12	13.13	14.14	15.15
8.12	8.84	9.48	10.10	10.74	11.37	12.00	12.63	13.89	15.15	16.42	17.68	18.96
9.85	10.61	11.37	12.12	12.88	13.64	14.40	15.15	16.67	18.18	19.70	21.22	22.74
11.50	12.38	13.27	14.14	15.03	15.92	16.80	17.68	19.45	21.21	22.98	24.75	26.54
13.13	14.14	15.15	16.16	17.17	18.18	19.19	20.20	22.22	24.24	26.26	28.28	30.30
14.78	15.91	17.05	18.18	19.32	20.46	21.59	22.73	25.00	27.27	29.55	31.82	34.10
16.42	17.68	18.94	20.20	21.47	22.73	24.49	25.25	27.78	30.30	32.83	35.35	37.88
18.07	19.45	20.84	22.22	23.62	25.01	26.89	27.78	30.56	33.33	36.11	38.89	41.68
19.70	21.21	22.73	24.24	25.74	27.27	28.79	30.30	33.33	36.36	39.39	42.42	45.45
20.35	22.98	24.63	26.26	27.89	29.55	31.19	32.73	36.11	39.39	42.68	45.94	49.26
22.98	24.75	26.51	28.28	30.10	31.82	33.58	35.35	38.89	42.42	45.96	49.49	53.03
24.63	26.47	28.41	30.30	32.15	34.10	35.98	37.87	41.67	45.45	49.24	53.03	56.82
26.26	28.28	30.30	32.32	34.34	36.36	38.38	40.40	44.44	48.48	52.52	56.56	60.60
27.91	30.05	32.20	34.34	36.49	38.64	40.78	42.93	47.22	51.51	55.80	60.09	64.40
29.54	31.82	34.09	36.36	38.64	40.92	43.18	45.45	50.00	54.54	58.09	63.63	68.18
31.19	33.59	35.99	38.38	40.79	43.20	45.58	47.98	52.78	57.57	62.37	67.16	71.98
32.83	35.35	37.87	40.40	42.93	45.45	48.98	50.50	55.55	60.60	65.65	70.70	75.75
39.39	42.42	45.45	48.48	51.51	54.54	57.57	60.60	66.66	72.72	78.78	84.84	90.90
45.95	49.49	53.02	56.56	60.19	63.63	67.16	70.70	77.77	84.84	91.91	98.98	106.05
52.52	56.56	60.60	64.64	68.68	72.72	76.76	80.80	88.88	96.96	105.04	113.12	121.20
59.09	63.63	68.18	72.72	77.27	81.81	86.36	90.90	100.00	109.08	118.17	127.26	136.35
65.65	70.70	75.75	80.80	85.85	90.90	95.95	101.01	111.11	121.21	131.31	141.41	151.51

interpolate. The values given include the factors required for conveyors shorter than 150-foot centers. In a similar way Table 18 gives values for $\frac{2.33\,TL}{33{,}000}$ for the material carried on conveyors of various capacity and length, including the factors required for conveyors shorter than 150-foot centers. For capacities or lengths not given in the table, interpolate.

For a level or horizontal conveyor, add the values taken from Table 17 and Table 18. For an inclined conveyor, add to the sum of these, the horse-power taken from Table 19 which is based on formula (3), page 97.

Horse-power of Conveyors with Improved Idlers.—Laboratory tests of troughed-belt idlers of various kinds show that the coefficient of friction falls off when the bearings are lubricated by oil, and that it is still lower when roller bearings or ball bearings are used. The comparative values are about as follows:

Coefficient of friction, grease-lubricated idlers..	100 per cent
Coefficient of friction, oil-lubricated idlers.....	60 per cent
Coefficient of friction, roller-bearing idlers.....	26 per cent

This does not mean that when a conveyor is equipped with oil-lubricated bearings, for instance, that the horse-power will be 60 per cent of that given by formula (2) or that $(100-26)=74$ per cent of power will be saved by using roller-bearing idlers. The reason is that the coefficients used in the derivation of that formula include, as has been stated on page 96, not only friction in the idler bearings, but also several other factors which are external to the idlers and which will remain in spite of improvements in the idlers.

There is no doubt that considerable power is saved in long conveyors by using idlers with ball or roller bearings, or even oil bearings, but there are no available records of direct comparative tests. Statements by manufacturers are given in Chapter IV, and some comparisons have been made between the actual horse-power required for conveyors equipped with ball or roller bearings and the horse-power which it is estimated these conveyors would have taken had they been fitted with grease-lubricated idlers. Several comparisons of that kind are given in Chapter IV.

If a horse-power formula were developed in which the idler losses were treated separately from the other losses—namely, bending the belt, speeding up material at the loading point, lifting the load over the idlers and the losses in the conveyor terminals, then it would be possible to say what power could be saved by better idlers. Unfortunately, however, there are no data available to establish such a formula.

An empirical formula which agrees well with tests of the three conveyors referred to on page 92, and with tests of a number of long conveyors 36 inches or more in width, fitted with ball-bearing idlers (p. 90), is

$$\text{h.p.}=\left(\frac{.0087\,L}{100}+\frac{.01\,H}{10}\right)T \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

This is in the form of formula (4). Compared with (4), it shows that the horse-power required for conveying, exclusive of the lift, is, with roller-bearing or ball-bearing idlers in good condition, a little less than half that required for plain troughing pulleys with grease lubrication.

Long-distance Belt Conveying—From formula (4) (p. 98) it is apparent that in a level belt conveyor where $H=0$, 1 h.p. will convey 1 ton per hour a distance of 5000 feet over plain grease-lubricated idlers, or, roughly, 1 h.p.-hour will convey 1 ton 1 mile. On the basis of power consumption alone a belt conveyor that carries 1 ton 1 mile in one hour is not so economical as a horse and cart, and for handling large quantities of material over long distances a train of cars drawn by a steam or electric locomotive will require much less power than a belt or a series of belts. To buy belts with their carrying idlers and driving machinery and build a structure to support the conveyor will generally cost more in such cases than to lay track and provide cars and motive power suited to the quantity of material to be handled.

There are, however, conditions in which a long belt system may cost so much less for operating labor and attendance than a car system that it will pay to install the belt conveyors. For example, an important mining company had under consideration for several years a plan to carry 8500 tons of coal per day from three inland mines to a shipping point on a river over four miles away. The choice lay between mine-car haulage with electric locomotives on the one hand and a belt conveyor system on the other.

From costs derived by the company from the operation of many haulage plants and several large belt conveyors, comparative estimates were made which showed: 1. The total cost of installation of the belt conveyor system was about 25 per cent below that of the haulage system, and the annual charge for interest was correspondingly less. 2. The annual charge for depreciation for the conveyor system was about 15 per cent higher; this was based on an estimated life of three years for belts, five years for idlers, fifteen years (average) for other machinery, thirty years for steel and concrete work. 3. The estimated power cost for the belts was double that for the haulage. 4. The probable cost of labor and attendance for the belt system was about half the corresponding charge for the car system. This one item showed a difference of about \$85,000 between the annual operating costs and when balanced with the other items mentioned, it was sufficient to show an estimated saving of over three cents per ton of coal carried.

As a result of this comparison it was decided to install the belt system, and the work is now under construction (1922). The distance from the loading station in the mine to the bin at the river is 23,500 feet, and the total rise is about 200 feet. The conveyors, 20 in number, will be entirely underground in existing mine passages which have been widened and straightened for the purpose. The idlers will be fitted with roller bearings or ball bearings. From power readings of similar conveyors installed at other mines it is expected that 1 h.p.-hour will convey $2\frac{1}{2}$ tons 1 mile. This

is more than twice the amount which, according to the usual formulas, can be carried over ordinary idlers with grease lubrication in the bores of the pulleys.

Future Development in Long-distance Conveying.—The merits of improved carrying idlers are described in Chapter IV. Briefly, they save power, reduce the cost of belts and require less attendance. These factors are worth considering even in small conveyors, but when large quantities of material are to be handled over long distances they are of the greatest importance. It is reasonable to expect great improvements in the construction and lubrication of ball-bearing and roller-bearing idlers; when this is brought about, and when improved loading means are used, it is probable that belts will be run faster than at present, possibly up to the speed at which fine material will be blown off the belt by the resistance of the air. Belt conveyors will then be more efficient machines for the transport of large quantities of material and can be used economically for distances which now seem impracticable.

Relation of Horse-power to Belt Tension.—Knowing the horse-power to drive a belt conveyor, we can find out the tension in the belt. The

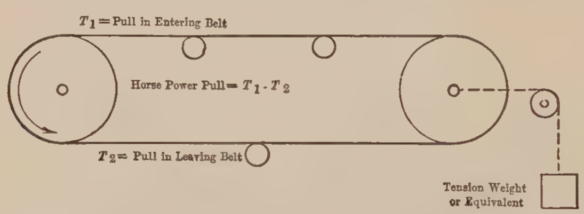


FIG. 98.—Relation between Horse-power Pull and Belt Tensions.

horse-power pull, or effective pull, is $\dfrac{\text{h.p.} \times 33{,}000}{\text{belt speed in ft. per min.}}$ in pounds. This is not the actual or total pull in the belt, because in order to maintain the belt in driving contact with the driving pulley the belt must be kept under tension on both the entering and the leaving sides of the pulley. If in Fig. 98 we call the pull on the entering belt T_1 and the pull in the leaving belt T_2, the effective pull in the belt which does useful work or which transmits horse-power is $T_1 - T_2$. This gives a difference of belt tensions but not the tensions themselves. From considerations of belt wrap and belt friction we can find the ratio of the tensions $\dfrac{T_1}{T_2}$, and by combining this with $T_1 - T_2$ we determine the actual values of T_1 and T_2.

Thickness of Belt as Determined by Belt Tension.—The total tension T_1 on the pulling side of a belt is made up of the horse-power pull plus an added tension necessary to maintain a driving contact between belt and pulley. The tension T_2 on the leaving side should be only what is necessary to maintain the driving contact. The ratio $\dfrac{T_1}{T_2}$ depends upon the

coefficient of friction between belt and pulley and on the angle in degrees of belt wrap on the pulley. Calling the former f and the latter a, the mathematical expression is

$$\frac{T_1}{T_2}=10^{.00758fa}\,[1] \qquad (6)$$

Experiments have shown that for dry, clean rubber belts on cast-iron pulleys $f=.25$, and when the pulleys are lagged or covered with rubber $f=.35$. Table 20 gives values of $\frac{T_1}{T_2}$ for these two coefficients and for various angles of belt wrap.

TABLE 20—RATIO OF $\frac{T_1}{T_2}$ FOR VARIOUS CONDITIONS OF DRIVING

T_1 = Pulling Tension at Drive Pulley, T_2 = Slack Tension at Drive Pulley

Angle of Belt Wrap, Degrees	Bare Iron Pulleys	Lagged or Covered Pulleys	Angle of Belt Wrap, Degrees	Bare Iron Pulleys	Lagged or Covered Pulleys
135	1.80	2.28	215	2.56	3.72
140	1.84	2.35	220	2.61	3.83
145	1.88	2.43	225	2.67	3.95
150	1.92	2.50	240	2.85	4.33
155	1.97	2.58	255	3.04	4.75
160	2.01	2.66	270	3.25	5.20
165	2.06	2.74	285	3.47	5.70
170	2.10	2.83	300	3.70	6.25
175	2.15	2.91	315	3.95	6.85
180	2.19	3.00	330	4.22	7.51
185	2.24	3.10	345	4.51	8.23
190	2.29	3.19	360	4.80	9.02
195	2.34	3.29	420	6.25	13.00
200	2.39	3.39	500	8.86	21.21
205	2.45	3.50	600	13.71	39.06
210	2.50	3.61	700	21.21	71.96

Since the torsion at the drive shaft is measured by the horse-power pull which is T_1-T_2, and since the total pull in the belt is T_1, the ratio $\frac{T_1}{T_1-T_2}$ expresses the relation between the actual effective horse-power pull and the total pull in the belt. These ratios are given in Table 21.

From Table 20 we see that for a simple drive with 180° wrap on an iron pulley $\frac{T_1}{T_2}=2.193$ or $T_2=.456T_1$; then since horse-power pull $=T_1-T_2$ horse-power pull $=.544T_1$ or $T_1=1.838$ horse-pull (see Table 21). That is, the total tension in the pulling side is over 1.8 times what is necessary to move the load and overcome idler friction.

If the pulley is lagged, the ratio falls to 1.5, and if by use of a snub pulley the wrap on the lagged pulley is increased to 240°, the total tension

[1] See Kent's M. E. Pocketbook.

T_1 becomes only 1.3 times the horse-power pull on effective tension. (See Table 21.)

TABLE 21.—RATIO OF $\frac{T_1}{T_1 - T_2}$ FOR VARIOUS CONDITIONS OF DRIVING

T_1 = Tension in Pulling Belt at Drive Pulley, T_2 = Tension in Slack Belt at Drive Pulley, $T_1 - T_2$ = Horse-power Pull

Angle of Belt Wrap, Degrees	Bare Iron Pulleys	Lagged or Covered Pulleys	Angle of Belt Wrap, Degrees	Bare Iron Pulleys	Lagged or Covered Pulleys
135	2.25	1.78	215	1.64	1.37
140	2.19	1.74	220	1.62	1.35
145	2.14	1.70	225	1.60	1.34
150	2.08	1.67	240	1.54	1.30
155	2.03	1.63	255	1.49	1.27
160	1.99	1.60	270	1.45	1.24
165	1.95	1.57	285	1.41	1.21
170	1.91	1.55	300	1.37	1.19
175	1.87	1.52	315	1.34	1.17
180	1.84	1.50	330	1.31	1.15
185	1.81	1.48	345	1.29	1.14
190	1.78	1.46	360	1.26	1.13
195	1.75	1.44	420	1.19	1.08
200	1.72	1.42	500	1.13	1.05
205	1.69	1.40	600	1.08	1.03
210	1.67	1.38	700	1.05	1.01

If the belt is wrapped around two driving pulleys as in the tandem-geared drive (Fig. 102, p. 121) the total angle of wrap may be greater than 360°, the ratio $\frac{T_1}{T_2}$ increases (see Table 20) and the ratio of T_1 to the horse-power pull decreases toward 1; when that point is reached, all of the strength of the belt is effective for moving the load and overcoming idler friction. Practically, the ratio never reaches unity, but for a wrap of 420°, the total tension is only 8 per cent more than the effective horse-power tension (see Table 21); this means that a belt for such a case need hardly be thicker than is required to transmit the horse-power pull, but while for a plain 180° wrap a 4-ply belt, for instance, might transmit the horse-power pull, it would take 4×1.8 (see above) $= 7.2$ plies to give the added strength necessary to maintain the driving contact between the belt and the pulley.

Calculation of Number of Plies.—The first step is to determine the horse-power of the conveyor from formula (4) on page 98 or from formula (5), page 106, or from Tables 17, 18, 19, then the horse-power pull from horse-power pull $= \frac{\text{h.p.} \times 33{,}000}{S}$ where S is belt speed in feet per minute. This horse-power pull multiplied by the factor from Table 21 gives the necessary stress T_1 on the pulling side of the belt.

Belts are said to be worth so many pounds per ply per inch of width, calling this figure p, and the number of plies n, and the width of belt W, then $n = \frac{T_1}{pW}$.

Ultimate Strength and Working Tension.—Haddock's experiments with a 12-inch 4-ply conveyor belt (Transactions, A. S. M. E., Vol. 30, 1908) showed a stretch which was not considered excessive under 1500-pound belt tension or 31 pounds per inch per ply. When the load was increased to 42 pounds per inch per ply the belt in its total length of 158 feet stretched 3 feet more than under the 31-pound tension. When the load was increased to 62 pounds per inch per ply there was a further stretch of 8 inches which did not increase after 30 minutes run under that load.

The proper working tension is determined first by the stretch, and, second, by the necessity of avoiding pulls too high for the metal fasteners generally used to join the ends of belts. (For fastenings, see Chapter III.) Baldwin in Marks' M. E. Handbook, 1st ed., p. 1179, gives 18.4 pounds per inch per ply as the proper tension; the line diagrams published by Robins are based on that same value; Stephens-Adamson recommend 20 pounds except for temporary work; Goodyear uses factors based on 27.5 pounds for belts made of 36-ounce duck, 25.2 pounds for 32-ounce duck and 23. pounds for 28-ounce duck; Goodrich uses 24 pounds for 32-ounce or 28 ounce duck; Jeffrey uses tables based on 30 pounds; Main Belting Co. uses 30 pounds for 32-ounce canvas belt duck (see p. 46).

There is no notable difference in the ducks used in the rubber belts made or sold by these various concerns; the differences in the allowed working tensions merely represent differences of opinion as to what is proper and safe. It is true, however, that there is a growing tendency toward the use of higher unit stresses than were common five or ten years ago.

Ultimate Strength of Belts.—These values are based, of course, on the ultimate strength of the belt. It has been the practice to say that ordinary conveyor belts would show an ultimate strength of 360 pounds per inch per ply; but since belts vary as to weight of duck it is better to say that the following represents an average for belts of American make when tested in the whole width: 28-ounce, 300 pounds; 30- and 32-ounce, 325 pounds; 36-ounce, 360 pounds. It is not usually possible to get these results from test strips 1 or 2 inches wide for the reasons given on page 39; such tests, as well as those of strips of duck either before or after making up into belt, may show considerable variation depending on the different methods of testing employed by manufacturers. On this point, see page 33. The weight of duck is in itself no sure indication of the strength of the belt; the degree of twist in the threads, and the number of threads in warp and filler, the method of vulcanization—all of these have their influence; but as a basis for estimating working strengths, the values given may be taken as representing average belt manufacture.

Working Tension of Belts.—In choosing the working tension it is well to be guided by the conditions under which the belt is to be used. If it

forms part of a permanent installation, has a cover thick enough for protection, and seems no more likely to fail from cutting and abrasion than from separation of plies or loss of " life " in the friction, then for long service the unit stress should be under 25 pounds; but for a temporary job, or where the life of the belt is likely to be terminated by an accident or by the loss of its cover, then the unit stress can be made higher than 25 pounds and up to 30 pounds.

For a practical example, where the belt tension was made higher than 25 pounds, see p. 117.

Faulty Methods.—In some line diagrams published to show proper belt thickness and in the usual formulas the unit belt stresses are not stated, but are included in factors or " constants " which vary according to the nature of the drive, the effect being to make the allowable unit stress high for the lagged drive with a large angle of wrap and low for the drive with a plain iron pulley with a wrap of 180° or thereabouts. These methods do not show any relation between the strength of the belt and the tension under which it is worked; it seems much better to start with a definite unit stress per inch per ply and fix the belt thickness after the total belt stress has been determined. This is the purpose of Table 20 and Table 21. From these the actual belt stresses T_1 and T_2 can be calculated and from them we can find the size of the drive shaft, the strength of the machinery supports and other details in the design of a belt conveyor. In the other methods referred to the true values of T_1 and T_2 are masked in the calculations and cannot readily be determined.

A formula published in 1908 (Trans., A. S. M. E., Vol. 30) and still retained in a few publications gives

$$X = \frac{\text{h.p.} \times 33{,}000}{SB};$$

where X = stress in the belt in pounds per inch of width;
S = belt speed in feet per minute;
B = width of belt in inches.

This is wrong because it ignores the difference between horse-power pull and T_1 (see p. 109). For a simple drive with 180° wrap it gives values too low by 45 per cent.

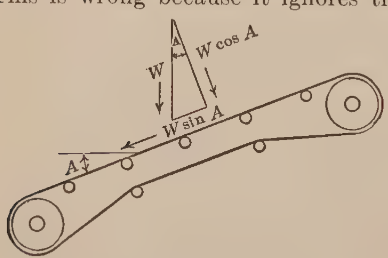


Fig. 99.—Belt Tension Due to Angle of Incline.

Tension in Belts due to Incline of Conveyor.—In an inclined belt conveyor, if W (Fig. 99) is the weight of the belt on each run, and A in the angle of incline from the horizontal, the component of W parallel to the incline directed downward is W sin A and the direct load on the idlers is W cos A. The

resistance of the latter to turning is $W \cos Af\frac{d}{D}$ or about $.07W \cos A$ with grease lubrication, and therefore the tendency of the belt to run downhill or produce an added tension at the head is $W(\sin A - .07 \cos A)$.

When $A = 4°$, $\sin A$ is nearly equal to $.07 \cos A$, and the expression becomes zero, that is, on angles less than 4° there is no tendency of the unloaded belt to run downhill and hence no belt tension at the head due to the angle of incline and the weight of the belt.

Table 22 gives values of $(\sin A - .07 \cos A)$ for various slopes. Thus, on a 20° incline 400 feet long with grease-lubricated idlers the pull in a belt weighing 10 pounds per linear foot is $400 \times 10 \times .28 = 1120$ pounds. Since this is exerted on each run of belt, the total pull due to the weight of the belt which produces bending in the head shaft is 2240 pounds.

TABLE 22.—PULL IN CONVEYOR BELT AT HEAD PULLEY DUE TO ANGLE OF SLOPE

Angle of incline A—Degrees	2°	4°	6°	8°	10°	12°	14°	16°	18°	20°	22°	24°	26°
	Percentage of Weight of Belt												
Grease lubricated idlers	0	0	3	7	10	14	17	21	24	28	31	34	37
Ball bearing or roller bearing Idlers	0	4	7	11	14	17	21	24	28	31	34	38	41

When ball-bearing or roller-bearing idlers are used the coefficient of idler friction may be as low as .03; then at all angles over 2° there is a tension in the belt due to the incline which increases to greater percentages of the weight at the angles of incline ordinarily used. These are given in the last line of the table.

Width of Belts.—The width of a belt depends, first, on the size of the pieces handled, and, second, on the capacity required. So far as capacity is concerned, the width can be determined from the speed and from Table 26, but the more important requirement is that the belt shall be wide enough for the largest pieces. This is really based on the necessity of having a loading chute that will not choke. A chute, to avoid spilling off the belt, can hardly be wider than $\frac{2}{3}W$ (W = width of belt), and since the width of a chute to avoid choking must be at least three times the size of the pieces when they are uniform in size, it follows that the belt width should be about five times the size of the pieces where they are all of one size, as in sized anthracite coal and other screened materials (see Table 23, column A). But run-of-mine coal, crushed rock and such material handled on belt conveyors seldom consists of pieces of uniform size. For these, the width of the chute can be about twice the size of the largest pieces with a

fair chance that no two of them will lock in position to block the flow. Under those conditions the ratio of belt width to the size of the occasional large lump can be according to column B of Table 23.

TABLE 23.—WIDTH OF BELT ACCORDING TO SIZE OF MATERIAL

Width of Belt, Inches	*A* Size, Inches	*B* Size, Inches	Width of Belt, Inches	*A* Size, Inches	*B* Size, Inches
12	1½	2	32	6½	12
14	2	3	34	7	13
16	2½	4	36	7½	14
18	3	5	38	7½	15
20	3½	6	40	8	16
22	4	7	42	8½	17
24	4½	8	44	9	18
26	5	9	48	10	20
28	5½	10	54	11	22
30	6	11	60	12	24

NOTE.—If the material is uniform in size use Column *A*.
Column *B* shows the size of lumps which may safely be carried if those lumps do not exceed 20 per cent of the mass, and the remainder is mostly small.

Thickness of Belt as Determined by Troughing.—When belts are too stiff to conform when empty to the contour of the troughing idlers they lose the guiding effect of the horizontal pulleys and are likely to be deflected from a straight course by the steering action of the inclined pulleys (see p. 79). This effect of the inclined pulleys depends upon the angle of their inclination; it was very noticeable with 45° and 35° troughing angles and it led to the use of the stepped-ply belt (see p. 16). More recently it prompted the general adoption of angles of 30° or less, and in current practice it limits the thickness of belts to what will conform to the contour of three-pulley idlers troughed 30° or less or five-pulley idlers with angles of 15° and 30°. Excessive thickness leads to the use of side-guide idlers at short intervals, skewed idlers, tilted idlers, all expendients to overcome the natural tendency of troughing idlers to "steer" a belt (see p. 79).

On the other hand, if belts are too thin they deflect too much under load; they are more likely to be pinched in the "wedge angle" between idler pulleys (see Fig. 36, p. 15); and if side-guide idlers are used, the edge of the belt may be turned back or folded over by pressing against them. Moreover, the belt may crack longitudinally from a lack of enough filler threads to withstand the repeated bending back and forth over troughing idlers.

Based upon these considerations, Table 24 has been prepared to show the maximum and minimum number of plies for different widths of rubber belt. The values given are a composite between those given by several authorities; those for belts wider than 30 inches lean toward plies heavier than it was once thought proper to use; but since those sizes are generally run on five-pulley idlers it is not necessary to limit them to those thicknesses

that will bend sharply. The natural bend of the heavy belts shown in Figs. 76 and 77 is much more pronounced than is necessary to fit any five-pulley standard idler. Fifty-four-inch belts with 11 plies and 60-inch belts with 12 plies of extra heavy duck are in successful use in five-pulley idlers.

TABLE 24.—MAXIMUM AND MINIMUM THICKNESS OF RUBBER BELT FOR PROPER TROUGHING AND LOADING ON STANDARD IDLERS

Width of Belt, Inches	Maximum Plies	Minimum Plies	Width of Belt, Inches	Maximum Plies	Minimum Plies
12	4	3	32	8	5
14	4	3	34	8	5
16	5	4	36	9	5
18	5	4	38	9	6
20	6	4	40	9	6
22	6	4	42	10	6
24	7	5	44	10	6
26	7	5	48	11	7
28	8	5	54	11	7
30	8	5	60	12	8

When belts are run flat or on flared idlers or on three-pulley idlers troughed 20° or less, the maximum thickness of belt is not limited to the plies given in Table 24, so far as troughing is concerned. Such idlers come nearer to matching the natural free bend of belts 24 inches and narrower than do five-pulley idlers and most commercial three-pulley idlers. (See Fig. 78.) For that reason they are better suited to thick rubber belts and especially to canvas belts and balata belts, which are naturally stiffer than rubber belts.

For thickness of belt as determined by the tension, see page 110.

Design of Belt Conveyors.—The successive steps in the design of a belt conveyor may be set down thus:

	Refer to
1. From known size of material select width of belt.	p. 114
2. If conveyor is inclined, assume a safe angle.	p. 142
3. Choose proper speed of belt according to nature of material, size of lumps, angle of incline, length of belt.	CHAP. VII
4. From 3 and from capacity required determine belt width. .	TABLE 26
5. Use 1 or 4, whichever is the larger.	
6. Consider what kind of supporting idlers are to be used. . . .	CHAP. IV
7. From 6 calculate horse-power for belt, and load, not including transmission losses. If conveyor is inclined or has a tripper add for that.	CHAP. V
8. From 7 and 3 calculate horse-power pull and from that the belt tension.	TABLE 21
9. Assume a unit stress per inch per ply and from 8 determine the number of plies.	p. 110
10. If the belt is too thick in proportion to its width, figure on a more efficient drive, or use a higher unit stress, or shallower troughing, or else use a wider belt.	p. 114

		Refer to
11.	Determine diameters of pulleys drive, foot, bend, snub or tripper	p. 127
12.	Lay out gearing to suit source of power.	—
13.	From 7 and 12, determine horse-power to drive by adding for losses in power transmission.	p. 98
14.	Calculate drive shaft for torsion and bending, other shafts for bending only.	—
15.	Consider kind of belt and, if rubber, thickness of cover. . . .	Chap. III
16.	Consider design of loading chute and skirt-boards.	Chap. VII
17.	Consider design of discharge chute.	Chap. VIII
18.	Consider location and style of take-ups.	Chap. VI
19.	Consider protective deck, cleaning brush, location of return idlers, use of tripper.	Chap. IX Chap. VIII

Errors or mistakes of judgment in the design of belt conveyors are not all equally serious. Some cause the belt to wear out sooner, and others can be corrected after the conveyor is in service. But there are two which are very serious; they cannot be corrected except after aggravating delay and much expense and often by humiliating makeshifts. These are a belt too narrow for the size of the material or the " peak " capacity, and an inclined belt too steep. Designers should be very careful on these points (see p. 148).

Design of a Specimen Belt Conveyor.—It was required to carry 2000 tons of limestone in ten hours from a crusher to a loading chute, length, 223 feet on 18° incline; height of lift, 69 feet. The crushed material weighed 100 pounds per cubic foot and ranged in size from 5 to 16 inches; average about 8 inches. The crusher was rated at 250 tons an hour, but it was fed rather intermittently from dump cars, and it could deliver at a higher rate for a few minutes when the rock did not require much reduction in size. It was decided to provide for an excess of one-third, or an hourly rate of 333 tons, equivalent to 6660 cubic feet per hour. A 30-inch belt would give that capacity at a comparatively slow speed (see Table 26), but on account of the 16-inch lumps it was necessary to use a 40-inch belt (see Table 23). This is usually rated at 5120 cubic feet per hour at 100 feet per minute (see Table 26), but since the conditions for loading the belt under the crusher were not favorable for a uniform feed it was taken at 4500 cubic feet. The belt speed for the capacity required was under 150 feet per minute, but after some consideration of motor speeds and room available for gearing it was decided to run the belt 225 feet per minute, but with a loading lighter than usual so as not to exceed 333 tons an hour capacity.

The next step was to determine the horse-power for the belt.

From Table 17 the h.p. to run the conveyor empty is $2.68 \times 2.25 =$	6.00
From Table 18 the h.p. to carry 333 tons over 223-foot level	= 5.25
From Table 19 the h.p. to lift 333 tons 69 feet vertically	= 23.20
or a total of 34.5 h.p.	34.45

The belt pull $T_1 - T_2$ corresponding to this is $\dfrac{34.5 \times 33{,}000}{225} = 5060$ pounds.

To determine the total tension T_1 in the belt refer to Table 21.

If we use a 180° wrap on a lagged pulley	$T_1 = 5060 \times 1.50 = 7590$	(No. 1)
If we use a 240° wrap with a snub pulley	$T_1 = 5060 \times 1.30 = 6578$	(No. 2)
If we use a 420° wrap with tandem pulleys	$T_1 = 5060 \times 1.08 = 5464$	(No. 3)

Rubber belts are rated anywhere from 18 to 30 pounds permissible tension per inch per ply (see p. 111). Taking 24 pounds, a 40-inch belt is worth 960 pounds per ply. On this basis, the first drive would take 8 plies; the second, 7; the third, 6 plies. We now make an assumption to get from Table 22 the pull due to the incline of the belt. A 40-inch 8-ply belt with $\frac{1}{8}$-inch cover weighs about 12.5 pounds per foot; the added tension for the 18° incline is $12.5 \times 223 \times .24 = 669$ pounds. Adding this, we get for the three cases 8259 pounds, 7247 pounds and 6133 pounds, for the total tension in the pulling side of the belt.

Of the 3 alternative drives, No. 1 requires more than 8 plies to keep the unit tension below 24 pounds; No. 2 would stress an 8-ply belt less than 23 pounds; while No. 3 would put only 22 pounds unit tension in a 7-ply belt with some saving in the cost of the belt.

It was decided to adopt No. 1, the drive with a half wrap on a lagged pulley and stress the belt to 26 pounds per inch per ply of 32-ounce duck. There were several reasons: it required less machinery at the head, and lighter supports; there would be no reverse bend in the belt and particles of stone would not be ground into the cover as would be the case if a snub pulley or a tandem-geared drive were used.

Sometimes a unit stress of 26 pounds would be open to objection, but in this case it was thought that the life of the belt would be fixed by the ability of the cover to withstand cutting and abrasion, and that a unit stress of 26 pounds might represent a proper balance for the life of the body of the belt considering the weight of the duck. The belt was installed with a 60-inch rubber-covered head pulley; it ran three years and handled a little over 2,000,000 tons of stone, a very satisfactory performance and quite remarkable, since the $\frac{1}{8}$-inch cover was rather light for the large pieces carried on the belt. The foot pulley was 42 inches in diameter, over 5 inches for every ply of belt, and the diameter of the head pulley was $7\frac{1}{2}$ inches for each ply. The good service of the belt may be credited in part to these large pulleys and to the fact that there were no reverse bends.

How *Not* to Do It.—The belt just referred to may be contrasted with another doing similar work in a different part of the country: a large rock crusher, fed by side-dump cars, delivered to a 36-inch belt inclined between 20° and 21°, 250-foot centers and run at 350 feet per minute. There was no feeder over the crusher nor between it and the belt. Some of the product was required to be in large pieces with least dimension about 8 inches, hence the crusher was set to produce that size. When the cars were dumped

the smaller stuff rushed through the crusher and on to the belt. On account of the high speed, 350 feet per minute, and the steep angle, the pick-up was bad, the rock did not acquire belt speed promptly and the sharp corners cut and tore the belt. Once started up the incline, pieces of rock would roll back and jump off the belt. To prevent that, continuous skirt-boards were then added for the length of the incline, and finally to prevent the tail end of a load on the belt from sliding or rolling back, sets of pawls or stops were placed over the belt every 20 or 30 feet. Each set consisted of eight 3-by-½-inch steel bars set edgewise, pivoted about 3 feet above the belt and mounted so that the lower ends of the bars could yield or move upward with the travel of rock up the incline but be rigid in the opposite direction to prevent any downward movement. Then a reciprocating plate feeder driven from the foot shaft was interposed between the crusher and the belt. When this installation was inspected it had been in service about six months, the cover of the belt showed signs of severe cutting and scraping, several sections had been cut out and replaced by new belt and belt fasteners were used to hold the belt together where it had been split lengthwise. The belt had been a good one, apparently 6-ply with a cover at least ⅛-inch thick.

The angle of incline in this case should have been 17° or less and the speed of the belt (see Table 30, p. 154) not over 200 feet per minute. This would have provided a capacity far in excess of the 1000 tons per day required and, with a feeder, would have allowed for some irregularity in the feed to the crusher without reducing the tonnage per hour.

Where to Drive.—The best place for a single-pulley drive is at the head or delivery end, because when the drive is placed there all of the loaded side is under tension, and it is less likely to run crooked than if it were partly slack. One way to make a belt run straight over troughing idlers is to pull it very tight (see p. 81); hence in conveyors driven at the foot there is a possibility that when the belt runs off to one side the attendant may help to straighten it by loading or adjusting the take-ups and thus put a stress in the belt far in excess of what is necessary for carrying the load or for maintaining belt contact on the driving pulley.

Driving at the Foot.—When an inclined belt is driven at the foot, the pulley contact there is reduced by the tendency of the belt to run downhill and form slack at the bottom. The force which acts in this way is on grease-lubricated idlers, $W(\sin A - .07 \cos A)$ (see p. 113). For a slope of 20° this amounts to 28 per cent of the weight of the belt on the up-run or on the down-run; hence, to maintain the proper relation between horse-power pull, T_1 and T_2 (see p. 110) the belt tension at the foot pulley must be increased by that amount by adjusting or loading the take-ups. The maximum belt tension in such a case is at the head, where it exceeds the tension at the foot by $W(\sin A - .07 \cos A)$.

Another difficulty with driving at the foot pulley is the disposal of the slack belt as it leaves the pulley. Usually the loading point is too close to the foot to allow any free hang of slack belt leading from the foot pulley. The take-up cannot engage the belt on the upper or loaded side; it is often

inconvenient to place it at the head because the discharge chute is there, and the alternative is to put it on the under side near the foot and let it take the belt under full tension.

When a belt conveyor is driven at the foot instead of the head, the pull on the head shaft and its bearings will be increased from $(T_1 + T_2)$ to $2T_1$. In spite of this and other drawbacks it is often advisable to drive the conveyor at the foot and then take care to keep the belt at the proper tension. Such cases are tailings stackers and similar conveyors where it is inconvenient to transmit power to the head shaft.

A Faulty Drive.—Fig. 110, page 129, shows a 36-inch coke conveyor driven at the foot pulley *B* with the take-up at *D*. The belt therefore made three half turns under full tension at *C*, *D*, *A*, and a bend at *E*. With the same general arrangement of pulleys, the conveyor might have been driven at *D* with the take-up at *C* or *B*. In this case the belt would have made only one-half turn under full tension at *A* besides the bend at *E*, there would have been less loss in journal friction and less wear on the belt, both internally and externally.

Multiple Pulley Drives.—From Table 21 it is apparent that the ability of a pulley to drive a belt increases rapidly when the angle of wrap exceeds 240°. It is seldom possible to get more than 200° on a plain pulley drive or more than 255° by using a snub pulley, but by using a second drive pulley engaging a reverse bend in the belt the combined angle of wrap may be made 360° or even more.

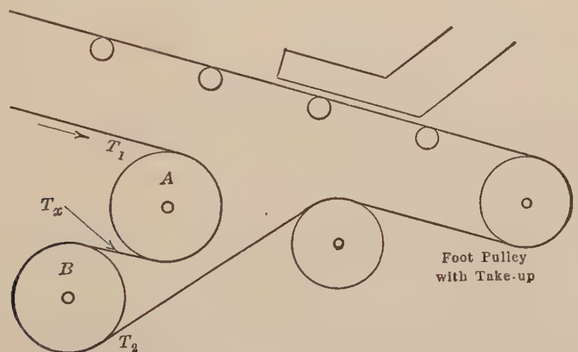


FIG. 100.—Belt Tensions in a Tandem-Pulley Drive.

Tandem Drives.—The action of tandem-pulley drive is shown in Fig. 100. If the pulleys *A* and *B* are of the same size and revolve at the same speed, the wrap on *A* being 180° and on *B* 240°, the following ratios exist when the pulleys are lagged with rubber: $\frac{T_1}{T_x} = 3.00$, $\frac{T_x}{T_2} = 4.33$. Therefore $T_1 = 3.00 \times 4.33\, T_2 = 13.00\, T_2$ (see Table 20). That is, for a comparatively low tension T_2 in the leaving belt it is possible to get a high driving tension T_1 in the entering belt and a high horse-power pull.

Comparison with Other Drives.—A single-pulley drive with 180° wrap on a lagged pulley requires a maximum tension T_1 of 1500 pounds for every 1000 pounds of effective horse-power pull; a snub drive with 240° wrap requires T_1 to be 1300 pounds, but in a tandem drive with a combined wrap of 420° on lagged pulleys T_1 is only 1080 pounds for the same effective pull of 1000 pounds. In most tandem drives the combined angle of wrap hardly exceeds 360°, for which T_1 would be 1130 pounds (see Table 21) for 1000 pounds effective horse-power pull. In general, a tandem drive will do with a given belt what requires a thicker and more costly belt on a single drive or a snub drive, or, stated differently, the tandem drive requires only a low initial tension in the belt for driving contact, while a single drive or a snub drive requires a higher belt tension. These ratios are given in Table 21.

In practice, the two pulleys are set close together on shafts connected by a pair of gears of equal size. For proper action, the intermediate tension T_x (Fig. 100) must be maintained, otherwise A will slip and fail to pull the load. This may happen if A is slightly larger than B; to prevent it, care must be taken that A and B are turned to exactly equal diameters, and if

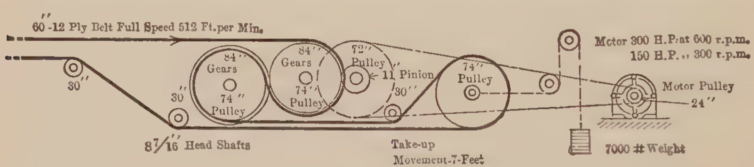


FIG. 101.—Tandem Drive for 60-inch 12-ply Belt. (Robins Conveying Belt Co.)

they are lagged, the lagging must be of the same thickness on both and kept so. Lagging is subject to wear, and if the covering of the second pulley B wears thin from contact with the dirty side of the belt, as happens when A is the head pulley (see Fig. 14), then the intermediate tension T_x may fall off and cause slippage. If, however, the lagging on A wears thin or comes off, the resultant slippage may hurt the belt. The Robins-Hersh patent of 1906 covers the idea of making the second driver slightly larger than the first, about $\frac{1}{4}$ inch on the diameter, so that the intermediate tension T_x is always maintained. This amount of difference means, with 36-inch drivers, about 2 or 3 feet slip per minute at ordinary conveyor speeds, not enough to do damage if the belt is clean, but apt to be injurious if the belt is dirty. (See p. 123 on the effect of slip and belt creep.)

Life of Tandem-driven Belts.—In general, belts used on tandem-driven conveyors do not last so long as those on single-pulley drives worked at the same tension. The chief cause is the reverse bend under load; secondary causes are injuries from accidental slip or local stretch between the drivers and from the normal amount of belt creep (see p. 123) always present in tandem drives.

Heaviest Belt-conveyor Drive.—Fig. 101 shows the arrangement of tandem-geared pulleys for a 60-inch 12-ply rubbe belt 2225 feet long,

driven by a 300-h.p. motor. It was installed in 1917 by Robins for the Baltimore and Ohio R. R. at a coal-shipping pier at Curtis Bay, Baltimore, Md., and is probably the heaviest belt-conveyor drive ever built.

Tandem Drives on Return Run.—It often happens on inclined belt conveyors or on long horizontal conveyors, that to drive at the head end means costly supports and placing the machinery in a dirty or unhandy place. A tandem drive on the return run near the foot puts the machinery in a cleaner place where it is more likely to receive proper attention because it is easily accessible. One drawback, not serious, is the added tension necessary to take care of the component of belt weight due to the incline (see p. 118). Other points to consider are the added cost of the driving machinery and bend pulleys as compared with a drive at the head, and the

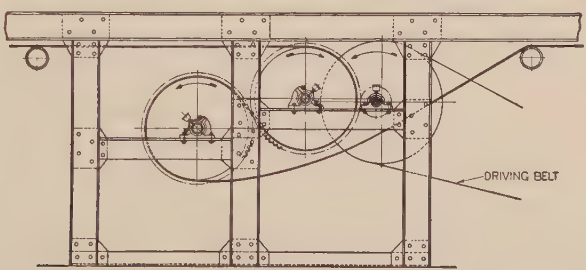


FIG. 102.—Tandem Drive on Return Run of Conveyor.

general objections to reverse bending and belt creep and slip (see p. 123). Fig. 102 shows a typical tandem drive on the return run of a conveyor.

For a tandem drive in which one of the pulleys is a snub pulley, see Fig. 109.

Lagging Tandem Pulleys.—From Table 20 it appears that for a combined wrap of 360° on two bare iron pulleys the ratio of $\frac{T_1}{T_2}$ is 4.80 and that for rubber-lagged pulleys the ratio is 9.02. This shows that for the same slack tension T_2 the driving tension in a conveyor belt can be nearly doubled by lagging the pulleys. For driving some long or heavily loaded belts it may be advisable to lag the pulleys to obtain a high pull with a comparatively slack belt leading from the second pulley (as the belt runs); but in other cases it is better to use bare pulleys and by means of take-ups, screw or weighted, make T_2 high enough to cause the bare pulleys to drive.

What this means in a practical case is stated in Table 25 which shows that in a large conveyor belt exerting 10,000 pounds effective horsepower pull, T_1 must be increased 1300 pounds and T_2 about the same amount, if the pulleys are bare and not lagged. This represents an increase of 10 or 12 per cent in the belt tension on the loaded side; it may not be objectionable if the unit stress in the belt is not excessive, or, in other words, if the pounds pull per inch per ply is within reasonable limits (see p. 111). Of course, means must be provided for giving the belt the necessary tension

T_2 as it leaves the second driver, preferably by a screw take-up or a weighted take-up placed near the driving group, as shown in Figs. 6, 8 and 14, and not at some remote point. There is an incidental advantage in having means for adjusting the slack tension in such drives; in cold weather when the belt is stiff and perhaps frosty, the coefficient of belt contact is low and the pulleys may slip in starting up for the day. If the slack tension is increased for a short time, the pulleys will take hold, and then, after the belt has gone around a few times and is in good working order, the tension can be reduced to the normal amount.

TABLE 25.—COMPARISON OF BELT TENSIONS IN TANDEM DRIVES

	Bare Iron Pulleys	Lagged Pulleys
Horse-power pull	10,000 lbs.	10,000 lbs.
Wrap in degrees	360°	360°
T_1 (See Table 21)	12,600 lbs.	11,300 lbs.
$\frac{T_1}{T_2}$ (See Table 20)	4.80	9.02
T_2	2,625 lbs.	1,250 lbs.
Increase of T_1	1,300 lbs.	
Increase of T_2	1,375 lbs.	

A Disadvantage of Bare Pulleys is that they wear out on the rim if the belt handles coke or similar sharp gritty material and if the belt is not carefully brushed clean. Bare tandem pulleys on coke conveyors have been known to wear out in less than a year, but when lagged pulleys were put in they did not have to be renewed. The lagging lasted eighteen months, but it is easier and cheaper to renew lagging than to replace pulleys.

Wear of Lagging.—Lagging the pulleys of tandem drives sometimes leads to trouble. When the lagging wears down, the driving diameter changes and it has happened that the second driver working against the dirty face of the belt wore smaller than the first driver, with consequent slip and a failure to drive. Unless the belt is brushed clean, fine dirt will be deposited on one of the drivers and may accumulate in thick patches. It is not easy to use a scraper (see 177) on a lagged pulley to remove such crusts and they may become so serious as to injure the belt or spoil the equality of belt speed on the two drivers. In some tandem drives the lagging has been removed from the pulleys for that reason, but this might not have been necessary if the belt had been brushed clean before its carrying side touched the driver.

FIG. 103.—Worn Bolt from Lagging of Conveyor Pulley.

Another difficulty encountered in some lagged tandem drives is that when the lagging wears thin, the bolts which hold it project and are then apt to cut and tear the conveyor belt. Fig. 103

shows such a bolt, which when the lagging wore away, projected, bent over, and then rubbed against the belt until half of the head was worn away.

The remedy for this trouble is to use pulleys with thick rims and with a large blunt-ended drill, remove the metal from around the bolt holes so that when the bolt is drawn tight, it will pull the lagging down into the countersink. Then the bolt head will be far enough below the outer surface to allow some wear of the lagging before the bolt head touches the belt (Fig. 104).

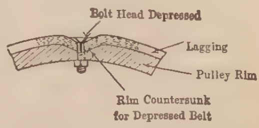


FIG. 104.—Countersinking Rim of Pulley for Lagging Bolts.

Belt Must Be Kept Clean.—On account of the movement between the pulley rims and the belt, caused by creep (not necessarily slip), it is very important to brush the belt well and keep it clean. It is cheaper to wear out brushes than to wear out lagging or the conveyor belt.

Tandem Drives: Wear of Pulleys and Lagging.—There are two factors that cause the wear of pulley rims and lagging referred to above, namely, **belt slip** and **belt creep.** Most persons acquainted with belts know what belt slip is; they have seen it. If the belt stands still and the pulley turns within it, the slip may be called 100 per cent of pulley travel because the travel of the belt has fallen to zero. When the pulley begins to drive the belt, the slip is less and when the drive is working properly, the slip is least. In single-pulley drives there is always some slip present (see p. 276), but in tandem drives the normal slip is probably very small.

Belt Creep is not so well understood because it cannot be seen. Chapter XX discusses creep in belt elevators, see Fig. 250; but in tandem drives for belt conveyors the creep is much greater because the ratio of tensions, instead of being 2 or 3 to 1, may be 9 or 10 to 1. A length of conveyor belt that measures 12.000 inches under no load may be stretched to 12.125 inches as it meets the first driver; but on leaving the second driver, the tension may be so low that the stretch is practically zero. This means that 12 inches of belt has shortened $\frac{1}{8}$ inch in passing through the drive, and if the belt speed is 400 feet per minute, the creep is 50 inches per minute and the wear on the pulley rim is represented by that much movement of two surfaces under severe pressure with particles of grit between them. Fifty inches per minute does not seem much, but it is the same as 140 miles per year of average service.

Belt creep is unavoidable; it exists in all belt conveyor drives. In plain drives on iron pulleys it is least because the ratio of tensions is least; as the pulling power of the drive increases, the creep increases also until it reaches a maximum in lagged tandem drives. It must be said, however, that the effect of creep is insignificant in most cases; it is only where the pull is severe and the belt dirty that the wear on pulley rims or pulley lagging can be called objectionable, even in a tandem drive.

Effect on the Belt.—So far as the belt is concerned, the wear from creep is spread over a surface so much greater than the surface of the pulley rim that it is seldom noticeable. Since creep is a result of belt stretch, it follows that a poorly made belt, or a belt too thin for the work, will creep more and cause more wear than a good belt on belt surfaces and pulley rims. When slip is combined with creep the damage is greater; this may happen when heavy conveyors, especially long inclines, are stopped and then started under full load, when the slack tension T_2 is too low to make the pulleys take hold, or when pulley rims are slippery from frost or condensed atmospheric moisture.

Driving by Pressure Belt.—Fig. 105 shows a method used by the Link-Belt Company for driving long and heavily loaded belt conveyors (Piez patent, 1919). The short pressure belt is maintained under tension by a suspended weight and wraps around the driving pulley outside of the conveyor belt. No power is applied to the pressure belt; it merely travels with the conveyor belt, upon which it exerts a tension depending upon the pull in the pressure belt due to the weight, and on the angle of contact between the two belts. The total tension on the pulling side of the conveyor belt is the sum of two items, one due to its own wrap on the driving pulley, the other due to the wrap of the short pressure belt. Both of these can be obtained from equation (6), page 109. If T_2 in Fig. 105 represents the slack tension in the conveyor belt, the first item T_1 equals $T_2 \times 10^{.00758fa}$, and if T_4 is the tension in the pressure belt where it leaves the conveyor belt, the second item which we may call T_3 equals $T_4 \times 10^{.00758fa}$. Then the total tension in the conveyor belt is $T_1 + T_3$ and if f, the coefficient of friction and a the angle of wrap are alike for both contacts, $T_1 + T_3 = (T_2 + T_4) \times 10^{.00758fa}$. These values can be obtained readily from Table 20.

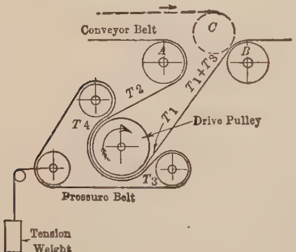


FIG. 105.—Pressure-belt Drive.

For a working example, suppose a conveyor belt has a wrap of 210° on a lagged pulley; then for an assumed tension of $T_2 = 100$ pounds on the leaving side, the greatest possible value of T_1 on the opposite side is $100 \times 3.61 = 361$ pounds for a single drive (see Table 20). If the pressure belt has a tension of 200 pounds due to the weight, and wraps the conveyor belt for an angle of 200°, then for a leaving tension $T_4 = 200$ pounds, the tension T_3 on the opposite side is $200 \times 3.39 = 678$ pounds (see Table 20). This added to $T_1 = 361$ pounds makes a total of 1039 pounds for the pull in the conveyor belt. The ratio of this to the original and unchanged slack tension of 100 pounds is 10.39, and from Table 20 it is seen that this is equivalent to the wrap of more than 360° and less than 420°; in other words, the device is equivalent, for the conditions stated, to a drive by two tandem-geared pulleys with a combined wrap of over 360°.

If, however, the tension in the pressure belt is raised to 500 pounds

by increasing the weight, then $T_3 = 500 \times 3.39$ pounds and $T_1 + T_3 = 2056$ pounds. The ratio of this to the assumed slack tension in the conveyor belt is 20.56, equivalent to a wrap of nearly 500° (Table 20). It is practically impossible to get such a high ratio of tensions with any arrangement of tandem-drive pulleys, and indeed such high ratios would be objectionable in causing an excessive amount of belt creep (see p. 123); nevertheless it is an advantage of the pressure-belt drive that it is quite easy to vary the pulling tension in the conveyor belt without disturbing the take-up tension in the conveyor. At the same time, the drive pulley works on the clean side of the belt and belt creep is not so likely to hurt the belt or the pulley as in tandem pulley drives.

Since it is possible to get such high ratios between slack tension and driving tension it is not necessary to lag the driving pulley, because the larger part of the driving effect can be obtained by the contact between the pressure belt and the conveyor belt, with less dependence on the contact between the latter and the driving pulley. For instance, if in the case cited above the pulley is bare, $T_1 = 2.50 \times 100 = 250$ pounds, then if the pressure belt is loaded to 300 pounds, $T_3 = 300 \times 3.39 = 1017$ pounds and $T_1 \times T_3 = 1267$ pounds. The ratio is then 12.67 and the driving effect is greater than that of a pair of lagged tandem pulleys with a combined wrap of 360°.

Fig. 106 shows a pressure belt drive applied to a 40-inch belt conveyor 630-foot centers, carrying 600 tons of coarse limestone at 300 feet per minute belt speed. It has an inclined part 230 feet long sloped at 13°, and the drive was placed just at the bend into the horizontal run. The drive pulley is 60 by 43 inches, not lagged; the pressure belt is 36 inches wide, 5-ply, rubber, loaded to 1500 pounds tension. The angle of wrap is about 195°, and for a leaving tension of 2000 pounds in the conveyor belt the pulling tension is about 9600 pounds.

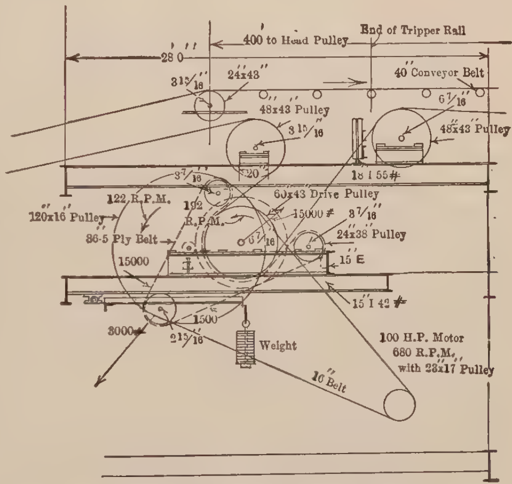


Fig. 106.—40-inch Conveyor Belt driven by Pressure Belt at Hump on Return Run.

Fig. 105 shows how the pressure-belt drive is applied to the return run of a belt conveyor with bend pulleys at A and B. It can be placed at the end of a conveyor, head or foot, by using a bend pulley at A with the head or foot pulley at C.

Page's Auxiliary Drive.—This device, patented in 1919, is shown in Fig. 107. It consists of a plurality of independently driven auxiliary belts in contact with the conveyor belt and supported by the conveyor idlers so that the driving tension is applied to the conveyor belt at several points. The idea is that the maximum tension in the conveyor belt will be less and that a very long belt can safely be made lighter and cheaper. No important installation of this device has been made (1921). The length of conveyor that an auxiliary belt can be expected to drive may be approximated in this way. Since the coefficient of friction between the two belts is not over .35 and the coefficient of journal friction for grease-lubricated idlers is, roughly, about 7 per cent of the weight of belt and load, then if W represents the weight of belt and load resting on a length of auxiliary belt L, the greatest pull which the auxiliary belt can exert is

Fig. 107.—Conveyor Belt with Auxiliary Drive Belts.

$.35W$, and that will drive $5.0L$ of conveyor belt beyond the auxiliary drive. For a particular case, the horse-power pull per foot of conveyor can be derived from one of the horse-power formulas; this should be used instead of the assumed 7 per cent of weight of belt and load in the approximation above. One limitation of this drive is that if the conveyor is loaded for only a part of its length, or if the load runs thin at intervals, the auxiliary drives under the bare places will exert little or no pull on the conveyor belt.

The Hoy patent 804474 of 1905 shows a method of driving a conveyor belt by means of an inner auxiliary belt when it is not convenient to apply power to either of the end pulleys of the conveyor.

Drive with Compensating Gearing.—The Hegeler and Holmes multiple-pulley drive, patented in 1916, consists of a compensating planetary gear device which distributes power to two or three driving pulleys in such a way that each pulley exerts an equal effort in driving the belt. The purpose of the gearing is the same as that of the differential gear in the axle of a motor car; in the conveyor drive it is intended to adjust the speed of each pulley for momentary stretch on the belt, for original differences in the diameters of the pulleys, and to allow for accidental differences in pulley diameters caused by damp sticky material forming a crust on the pulley rim. Fig. 108 shows such a three-pulley drive with pulleys 1, 2, 3 and a combined angle of wrap of 634°, designed for a belt 525-foot centers with a rise of 180 feet on a

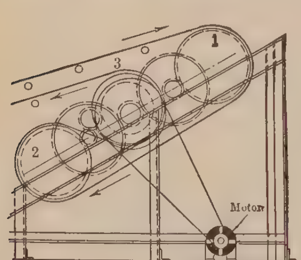


Fig. 108.—Three-pulley Drive with Compensating Gearing.

20° slope. No drive with this device has been built; it is certain that the advantages gained do not justify the expense and the complication of the gearing.

Three-pulley Drives.—The three-pulley drive has apparently some advantage in theory, but it has never been used; the main reason is that with an ordinary two-pulley drive it is only necessary to apply some tension to the leaving belt to get the desired tension in the entering belt. For instance, the ratio of $\frac{T_1}{T_2}$ for an angle of 634° is 15.9 for bare iron pulleys, and 48.09 for rubber-lagged pulleys. Taking the former ratio, a horsepower pull of 10,000 pounds means that T_1 is 10,700 pounds and T_2 is $\frac{10,700}{15.9}=673$ pounds, that is, the leaving belt is practically slack, with no tension. If we use two lagged pulleys with a combined wrap of 360° we avoid the complication of the third pulley with its gearing, and the belt creep inseparable from a high ratio of $\frac{T_1}{T_2}$. Then from Table 21, T_1 is 11,300 pounds and T_2 is 1250 pounds. That is, by increasing the belt tension only 600 pounds, we can make two pulleys do the work, and do it better.

Size of Pulleys.—In the formula on page 109, $\frac{T_1}{T_2}=10^{.00758fa}$, the only variables which determine the tractive force are the angle of wrap and the coefficient of friction; the diameter of the pulley does not enter into the calculation. Some have thought that a large pulley gets a better grip on a belt than a smaller pulley, but within the limits of good practice that is not true.

Haddock's experiments (Transactions, A. S. M. E., 1908) with a 12-inch 4-ply rubber belt showed no variation in tractive force whether it was driven by a 42-inch pulley or a 20-inch pulley, but on a 12-inch pulley the traction dropped 20 per cent. That was because the 4-ply belt was too stiff to bend to the 12-inch diameter without loss of contact pressure. These experiments established the rule that the diameter of the drive pulley should be at least five times the number of plies in the belt. When a pulley acts as a guide or a deflector, as at a foot shaft or a snub shaft, it is considered proper to make the diameter three or four times the number of plies, but, in any case, the larger the better.

The objection to making the ratio larger than 5 to 1 for drivers is that the pulleys take up more room and cost more, and more gearing or larger gearing is required because the larger pulley makes fewer turns for the same conveyor speed. On the other hand, a diameter larger than that given by the 5-to-1 ratio stresses the friction between the plies less, makes it less liable to crack when old and hence postpones the day when the belt fails by separation of the plies. (For an instance, see p. 117.)

An incidental advantage of a large head pulley is that when it is used

with a snub pulley the angle of belt wrap can be made large without having the snub pulley too small.

When the ratio for a driver is less than 4 to 1 the belt tension must be increased to make the belt hug the pulley tight enough to drive. This in itself is a disadvantage, but the greater harm is the excessive stretch in the friction layers between the plies and the greater tendency for the plies to come apart when the friction gets old.

Pulley Rims.—All drive pulleys, foot pulleys and snub pulleys should be specified as "double-belt" pulleys with a "crown" on the pulley face of at least $\frac{1}{8}$ inch per foot of face. A heavy crown will keep the belt centered even if the shaft on which the pulley is mounted should get out of level or out of square with the belt; hence some designers specify $\frac{3}{16}$- or $\frac{1}{4}$-inch crown per foot of face for drive pulleys where the belt might tend to get out of line. Pulleys wider than 24 or 30 inches should have double arms. The faces should be 2 inches wider than the belts up to 18 inches, 3 inches more for 24- or 30-inch belts and 4 inches more than belt width in the wider belts. This excess of width permits the belt to run out of center for a few inches without requiring it to be "trained" back into position by adjusting the troughing idlers or forcing it back by the use of edge rolls. It is quite necessary where the conveyor head or foot is supported on a frame which may settle or get out of line or where the conveyor runway is not permanently aligned. A few inches of excess width on the troughing and return idlers is also worth having under such conditions. It is better to incur that expense than to ruin a belt by the use of side-guide idlers or to damage the pulley side by skewing the troughing rolls to an excessive angle. For very heavy pulls, the rims of drive pulleys should have inside flanges for reinforcement.

Snub Pulleys.—If a conveyor belt makes a wrap of 180° on an iron pulley the tension T_2 in the empty belt must be $\frac{1}{2.19} = .456$ of that on the loaded side (see Table 20, p. 109). If the useful pull or horse-power pull is 2000 pounds, the total tension T_1 on the tight or loaded side must be $2000 \times 1.84 = 3680$ pounds (see Table 21, p. 110), and therefore the pull on the empty side is $3680 \times .456 = 1680$ pounds. But if a snub pulley (see Fig. 11) is used to increase the angle of wrap to 240°, the total belt tension T_1 is, for the same work, 3100 pounds, and the pull on the empty side T_2 is 1100 pounds, a reduction of nearly 600 pounds in the belt tension, equivalent to 1 ply of a 24-inch belt. This is the advantage of using a snub pulley; to offset it, there is the expense of the pulley and the reverse bending of the belt, also the tendency for sharp particles adhering to the belt to be forced into the cover by contact with the snub pulley.

For most horizontal conveyors handling coal a snub pulley is not required, and in other conveyors handling heavier materials it may be better to work the belt at a tension a little higher than normal rather than install a snub pulley. (For an example, see page 117.)

Sometimes the thickness of the belt will be made greater than is necessary

to transmit the pull; then it may be able to bear the tension necessary to get driving contact without the use of a snub pulley. (For reasons for increasing the number of plies, see Table 24 and p. 114.)

On conveyors handling materials which stick to the belt, a snub pulley is apt to receive a crust or layer which may build up so as to be objectionable. A steel scraper bearing against the face of the pulley will prevent it (see Figs. 177 and 178).

It is possible to gear a snub shaft to a head shaft and make the snub pulley help to drive the belt. Such a drive is shown in Fig. 109, which illustrates the arrangement at the head of a number of package conveyors in the Chicago Railway Post Office Terminal (1922). This gives a driving wrap of 160° on the snub pulley and 240° on the head pulley. It is in effect a tandem drive with the speeds of the shafts in the inverse ratio of their pulley diameters. The belts are stitched canvas, 4 or 5 ply, 36 to 48 inches wide, of 32 ounce duck.

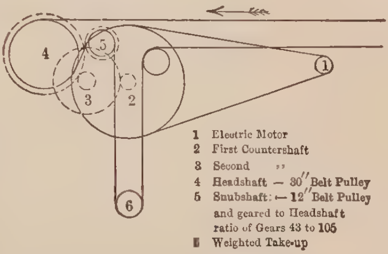


Fig. 109.—Snub-shaft Geared to Head Shaft and Acting as Second Pulley of Tandem Drive. (Lamson Company.)

Deflector pulleys around which the conveyor pull is transmitted should be as large as drive pulleys, that is, 5-inch diameter per ply of belt. Other pulleys can be 3 or 4 inches per ply of belt. An instance where the total pull is transmitted around deflector pulleys is shown in Fig. 110, representing an inclined coke conveyor driven at the foot B. Instead of making the head and foot pulleys A and B 36 inches and the deflector and take-up pulleys C and D 24 inches it would have been better to make all 36 inches, or at least all 30 inches because the belt was under practically the same tension in passing around all of them. The deflector E was 16 inches in diameter, but 24 inches would have been better for a similar reason. The belt in this case was 36 inches wide, 5 plies of 28-ounce duck, although the work required only 2 plies to take care of the maximum tension T_1. It lasted fourteen months and carried 240,000 tons of crushed coke at a cost of one-ninth of a cent per ton, a fairly good record; but if the pulleys had been larger it might have lasted longer than it did. Its fourteen months of service were not enough to make it die of old age. (For a suggested improvement, see page 119.)

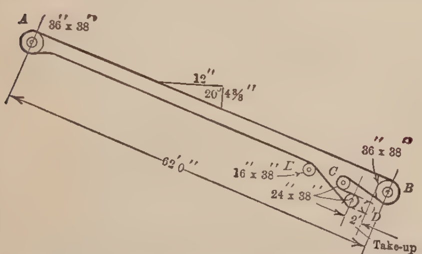


Fig. 110.—Conveyor with Deflector Pulleys too Small in Diameter.

Curves in Belt Conveyors.—Fig. 111 represents a belt conveyor AD, partly inclined and partly horizontal or at two different angles of inclina-

tion; the problem is to find the radius EB of the curve BC joining the two straight portions, which is large enough to prevent the belt from lifting off the idlers in the section BC under the most unfavorable condition, namely, when the section AB is fully loaded and the rest of the conveyor is empty. The curve BC is part of a catenary whose equation is $y = \frac{x^2}{2a}$ (Marks' M. E. Handbook, 1st ed., p. 148) where $a = \frac{T}{W}$ and T is the tension at B in pounds and W is the weight per foot of the empty belt in pounds. Therefore $y = \frac{Wx^2}{2T}$. The angle A of the inclined portion CD is determined by tang $A = \frac{2y}{x} = \frac{W}{T} x$ and the radius of curvature is $EB = \frac{T}{W}$ since the flat catenary is nearly a parabola.

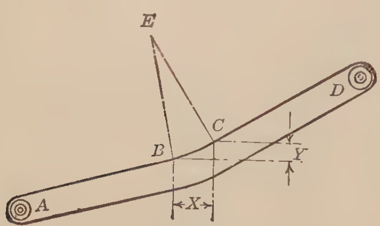


FIG. 111.—Radius of Curvature for Bends in Belt Conveyors.

Values of W are given on in Tables 4, 5, 6, 7, 8. To estimate T_1 the pull at B, consider AB as a loaded conveyor, and calculate the horse-power and then the horse-power pull corresponding to the belt speed (see p. 108). Then according to the method of driving the conveyor at D or elsewhere, multiply this horse-power pull by the proper factor from Table 21, page 110. If the inclination of CD is less than 4°, add for the pull on the return side of the incline 7 per cent of the weight of the belt on the lower run of the incline (see Table 22, p. 113). To the radius $\frac{T}{W}$ add something to provide for the chance that the take-up tension may be increased at times and the belt may be more likely to lift off the idlers at the curve. This tension may be increased accidentally by careless use of the take-ups or intentionally to give the necessary slack tension at the head to overcome slippage of the belt or to provide for extra heavy loading of the belt.

If such a conveyor is driven at A or at some place on the return side near A, the curve for the lower run must be calculated from the total horse-power pull for the conveyor, or it may be more convenient to run the belt under a bend pulley between two straight runs on the lower side and curve the upper run only.

Bends over Pulleys.—Where pulleys are used to change the direction of travel of belts under working tension they should be double-belt straight-face pulleys at least 4 inches in diameter for each ply of belt. (For the use of troughing pulleys at humps, see p. 74.)

CHAPTER VI

TENSION AND TAKE-UP DEVICES

Take-ups for Belt Conveyors.—A take-up does two things; it removes the accumulation of slack in the belt and permits the belt to be stressed to the tension at which the pulley will drive it. If the take-up is of the weighted or automatic type (Fig. 112) it must carry enough weight to load the empty belt to the tension T_2 at which the pulley can pull the load; thus, if a belt driven by a wrap of 210° on a lagged pulley has a total tension T_1 of 2000 pounds on the pulling side, the idle tension T_2 is $\frac{2000}{.361}=550$ pounds (see Table 20, p. 109) and when arranged like Fig. 112 the suspended weight of the pulley and its attachments must be 1100 pounds to give the belt the necessary tension. By the same reasoning, if the weight is applied to a foot-take-up, it must be twice the amount of the idle tension T_2.

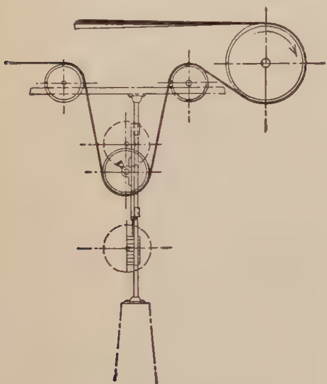


FIG. 112.—" Gravity " Take-up for Belt Conveyor.

A suspended or " gravity " take-up like Fig. 112 can be placed anywhere on the return belt. It requires little attention and keeps an even tension in the belt. The objection to it is that it gives the belt three extra bends.

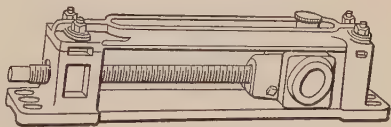


FIG. 113.—Screw Take-up.

A screw take-up (Fig. 113) is simpler and cheaper and when placed at the foot of the conveyor it requires no extra bends in the belt. In the hands of a careless man, a screw take-up may pull a belt much tighter than is necessary for driving contact, and thereby injure it or pull the lacing apart. On the other hand, if the accumulation of stretch is not removed as it forms, the slack tension at the driver may become too low; then when the belt slips, it may be scraped or torn on the pulley side.

In a tandem pulley drive (Fig. 102) or a drive with a pressure belt (Fig. 105) the ratio of $\frac{T_1}{T_2}$ is high, and for a given horse-power pull T_2 is comparatively low (see Table 25, p. 122); hence the belt coming from such

a drive may be under very low tension. When such drives are near the foot of a conveyor it may be convenient to let the belt hang slack between the driver and the foot; then with screw take-ups at the foot it is possible to maintain an even tension in the belt by keeping the sag uniform. A look at the hang of the belt will show whether it is at the right tension or not.

In general, the best place for a take-up is where the belt tension is low. One disadvantage of driving a conveyor at the foot pulley is the difficulty of disposing of the slack belt. (For an instance of this, see Fig. 110, p. 129.)

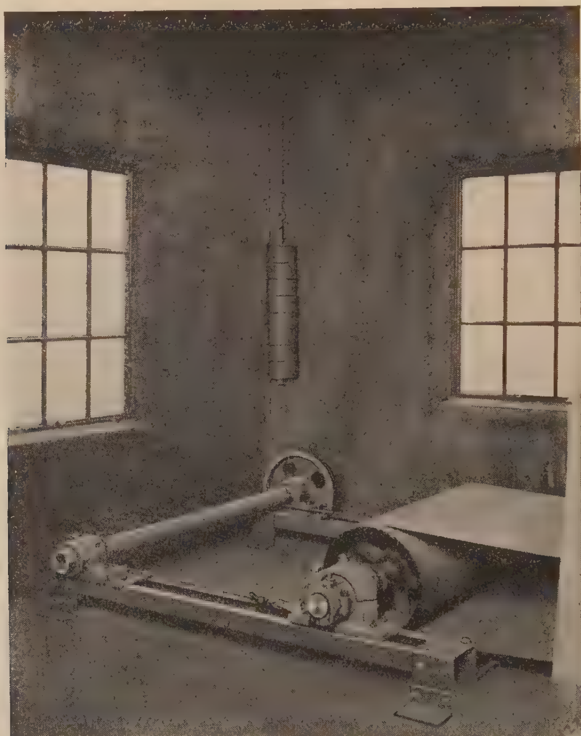


Fig. 114.—Weighted Pull Take-up for Grain Belt. (James Stewart & Co., Inc.)

Weighted Pull Take-ups.—Fig. 101, page 120, shows a take-up for a 60-inch 12-ply belt placed next to the driving end of a conveyor over 1000-foot centers.

Fig. 114 shows the foot of a grain conveyor with an automatic take-up, The sheave in the corner is mounted on a pipe shaft to which are fastened two ropes leading to the shaft bearings. The weight rope makes a number of turns around the sheave, and as the weight descends, the take-up ropes are wound upon the pipe under a tension equal to five or six times the weight.

CHAPTER VII

LOADING THE BELT

Loading Chutes.—Besides the normal duty of conveying material, a belt has to check the impact of material at the feed point and impart to it the belt's own velocity in the direction of travel. If 200 pounds of material falls from a height of 5 feet on a belt, the belt must absorb within its structure 1000 foot-pounds of energy in stopping its fall. It does it through the elasticity of its body of fabric and of its cover, if it has a cover. If there are hard sharp lumps, some of the energy of the falling mass is expended in cutting the belt or its cover, especially if the belt is so supported that it cannot exert its elasticity. This happens if an idler is placed directly under the point of impact. To avoid it, the arrangement should be like Fig. 115. The idler A avoids the direct impact, yet it prevents the belt from sagging too far from the skirt-boards B. If the belt were fed midway between two idlers, there might be too much deflection of the belt from impact, with consequent leakage sideways under the skirt-boards. If the idler is directly under the point of impact, there is the added risk of breaking the idler, see also Fig. 73.

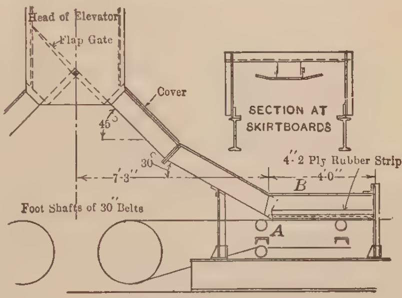


Fig. 115.—Loading Chute and Skirt Boards.

If, in the case referred to above, the conveyor runs 5 feet per second, the work it does in bringing the 200 pounds from zero velocity to 300 feet per minute is $\frac{1}{2}\times\frac{200}{32.2}5^2=78$ foot-pounds, and if the 200 pounds is fed on in two seconds, the equivalent is $\frac{78\times60}{2\times33,000}=.07$ h.p. for the capacity of 180 tons per hour. While this represents only a small addition to the normal pull in the belt, its effect on the carrying surface can be understood if we imagine a grinding wheel made of the material carried, pressed against the belt and driven by .07 h.p. for every minute the conveyor runs. Of course, some of the material falls on itself and does not touch the belt, but the example will serve to emphasize the statement that the arrangement of the feed is a very important item in the design of the conveyor and that a poor feed may spoil a good belt.

Proper Angles for Loading Chutes.—If belts could be fed with material moving at belt velocity (Fig. 116), cutting and abrasion would be at a minimum; but it is seldom possible in practice. The best that can be done is to deliver material through a chute pointing in the direction of belt travel and at such an angle that the horizontal component of the velocity in the chute will be equal to belt speed. If, as in Fig. 117, the velocity of the belt is V, the theoretical belt velocity of material in the chute is V secant A; the horizontal component of this is V secant A cosine A which $= V$, and the vertical component which is a measure of the impact on the belt is V secant A sine A. In practice, the best angle of chute can be determined only by trial; grain flows easily in chutes sloped 6 in 12 (27°) and rapidly at 8 in 12 (34°); but in Westmacott & Lyster's experiments in 1865 (see p. 7) it appeared that the best angle for delivering wheat to a belt moving at 500 feet per minute was 42½°. Anthracite coal in the domestic sizes flows readily in steel chutes at 5½ in 12 (25°), and in the steam sizes at 7 in 12 (30°), crushed soft coal requires

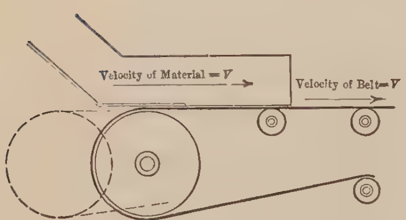


Fig. 116.—Theoretical Feed with Minimum of Abrasion.

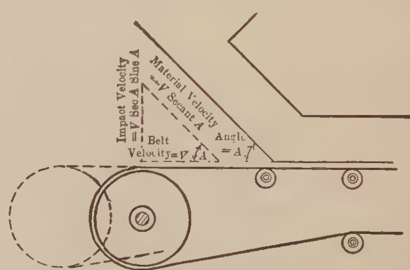


Fig. 117.—Relation of Belt Velocity to Flow in Chute.

8½ in 12 (35°). Screened and sized stone or ore will flow in steel chutes at 7 in 12 (30°), the same material mixed with fine stuff requires 8½ in 21 (35°). For all of these the slope of the loading chute should be 5° or 10° steeper, but for large lump coal or ore the angle should not exceed 45° (12 in 12); beyond that the lumps will not slide on the bottom of the chute, but descend in a series of jumps. Another factor with lumpy or sharp material is that in a steep chute the vertical component of its velocity becomes too great and the belt may be cut by the impact. To prevent that, the chute may be curved as in Fig. 123, or if the chute is long enough to give the material the necessary momentum, the angle may be broken, as in Fig. 115.

Both of these methods are effective in preventing, in some degree, the effect of impact and in giving the material some velocity in the direction of belt travel.

A form of chute used in loading run-of-mine coal on a 60-inch belt is shown in Fig. 118. The rounded bottom does not interfere with the flow of fine coal or pieces of moderate size, but lumps of very large size are apt

to be directed forward and upward by the converging corners so that they will strike the belt at B after it has already received a layer of the smaller coal at A. Another method of doing the same thing is shown in Fig. 119.

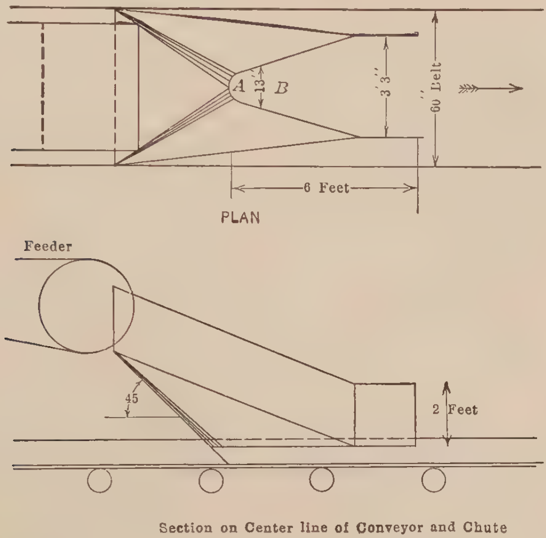


FIG. 118.—Belt Loading Chute for Run-of-Mine Coal.

Screen Chutes.—From the earliest days of the business, efforts have been made to reduce the wear on the belt surface by screening out the fines in the loading chute and delivering them to the belt first, so as to cushion the fall of the lumps. Fig. 120 shows the idea, but it is not easy to avoid choking such a chute; even if bars are used instead of perforated plate, pieces stick between them and the chute chokes or else the velocity of flow through it is reduced. Another construction is to put finger bars on the end of the chute and let the fines fall between, while the lumps ride over, but this is open to the danger of pieces catching between the bars or under them and dragging on the belt.

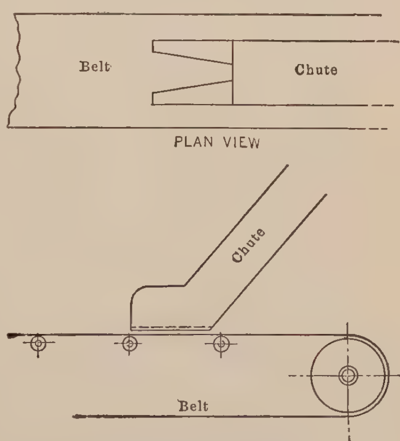


FIG. 119.—Loading Chute Designed to Save a Belt from Impact of Large Lumps.

When a belt is inclined at an angle close to 20° it is not easy to deliver material to it at belt speed; the lumps tumble around longer before becoming settled and the wear on the belt surface is greater. In such cases a

screen chute is a good thing if it is properly made. Figs. 121 and 122 show one designed to transfer run-of-mine coal from an apron feeder to a 36-inch belt inclined at 20°. The screening surface was about 30 by 50 inches, the bars were 4 inches by ½ inch spaced 3 inches; the spacers between the bars were set low to keep clear of lumps riding over the bars and the apron plate *A* was set below the top surface of the bars for the same reason. Since the bars did not extend close to the belt, there was no danger that lumps caught between the bars would drag on the belt and injure it. The main discharge from the apron feeder fell on the screen bars while the dribble under the head wheel was caught in the side extension of the chute and put on the belt behind the screen bars.

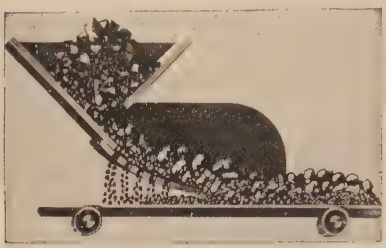


FIG. 120.—Typical Screen Chute.

Fig. 123 shows a similar chute without the screen bars but with a curved bottom designed to catch the lumps falling over the head of an apron

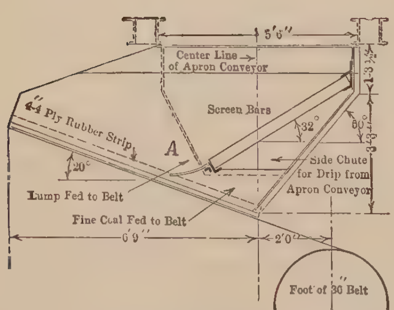


FIG. 121.—Screen Chute for Loading an Inclined Belt Conveyor.

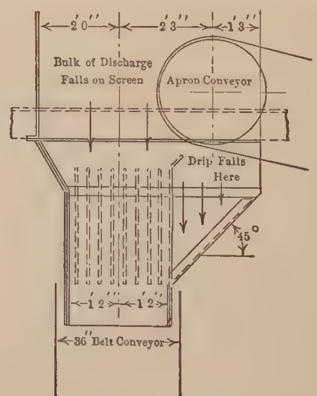


FIG. 122.—End View of Screen Chute Shown in Fig. 121.

feeder and deliver them to a 36-inch belt at belt speed. Some of the discharge from the apron conveyor, including the drip, falls into the side extension chute and is put on to the belt before the lumps so that the large pieces fall on a cushion of small coal.

Transfer Chutes.—In designing chutes to transfer from one conveyor to another, care should be taken that the discharge from the first belt cannot fall clear through on to the second belt without being guided by the chute, and also that the material is loaded *with* the run of the belt and not sideways. A belt loaded from the side will suffer from cutting and abrasion;

it is apt to leak under the skirt-boards, and with an unsymmetrical cross-section of load it will run crooked and need the assistance of side-guide idlers to keep it in place.

Skirt-boards.—When a horizontal conveyor is loaded by a properly designed chute it should not be necessary to extend the sides of the chute more than 3 or 4 feet to confine the material while it is coming up to belt speed and to prevent lumps from rolling off. These extensions should be kept a few inches above the surface of the belt and the space closed by a strip of rubber belt to prevent fine stuff from scattering and lumps from getting under the edge of the skirt-boards and jamming there. With steel skirt-boards the risk of jamming is less than with wood. In loading run-of-mine or crushed coal onto belts 36 inches or wider, where the width between chute sides or skirt-boards is not over $\frac{2}{3}W$ (W = width of belt), the rubber strips can often be omitted, because then there is margin enough between the chute and the edge of the belt to prevent loss of material. Fig. 124 shows a typical cross-section of wooden skirt-boards. Fig. 115 gives a design in steel construction. The chute shown in Fig. 118 has no skirt-boards or rubber strips.

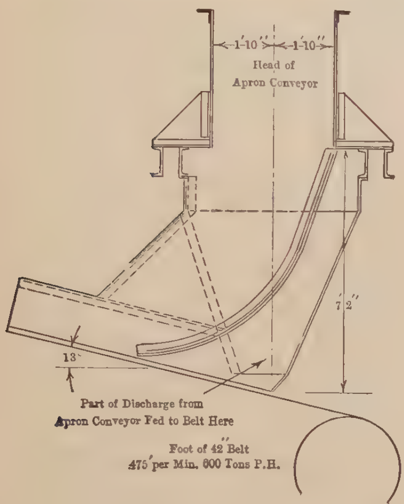


FIG. 123.—Curved-bottom Chute for Loading an Inclined Belt Conveyor.

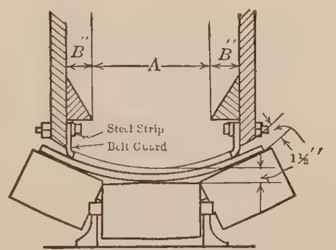


FIG. 124.—Wooden Skirt-boards with Rubber Belt Guard Strips.

Damage from Skirt-boards.—Nearly everyone in the belt conveyor business can think of cases where belts have been split or seriously injured by skirt-boards or chutes getting out of adjustment. To lessen the risk, skirt-boards should be firmly supported. They should be no longer than is necessary and there should be clearance enough under them to avoid cutting or scraping the belt if the latter, when running empty, should not lie down properly on the horizontal pulleys of the idlers. Means should also be used to prevent trippers from coming so close to the chute as to lift the belt under it (see p. 169).

Skirt-boards for Inclined Conveyors.—When the angle of inclination is over 10° the material does not acquire belt speed so readily, and lumps roll around more before becoming settled. For that reason, it is generally

necessary to make skirt-boards longer than for horizontal conveyors. If the incline is merely a short portion of the conveyor depressed to limit the travel of the tripper (see p. 170) the boards are sometimes extended to the hump (Fig. 133) to prevent loss of material there when the belt flattens out in going over the flat-faced pulley. But unless the sides are made continuous with those at the loading point, it is not safe to use skirt-boards at such humps, because material is sure to catch under them. When belts are loaded to their standard ratings (see Figs. 133 and 137) there should be no spill in passing over hump pulleys even though some of the material does describe a short trajectory and rise from the belt just as it passes the bend. In any case, it is better to reduce the load cross-section than to use separate skirt-boards at humps.

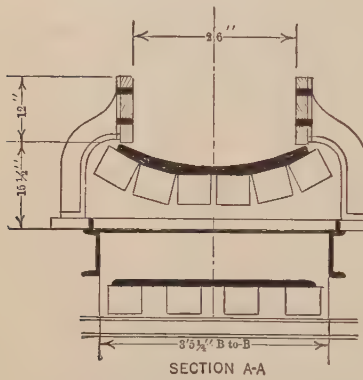


FIG. 125.—Continuous Skirt-boards for an Inclined Belt Conveyor.

In order to remove all danger of lumps rolling off the belt, skirt-boards for steep inclines are sometimes run the full length of the conveyor. Fig. 125 shows continuous guards of 2-by-12-inch planks for a 36-inch conveyor, 180 feet long inclined 20°. The bottom edges of the guards should be set above the belt so that if the belt shifts to one side it cannot rub against them nor their supports. For continuous guards on a grain belt, see Fig. 141.

In feeding dry, dusty material to a belt it is sometimes advisable to extend the skirt-boards and put a cover over them to make a box long

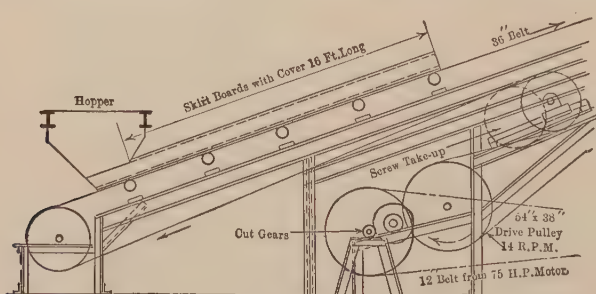


FIG. 126.—Covered Skirt-boards for Dusty Material.

enough to confine the dust. Fig. 126 shows such a guard for a 36-inch belt handling pulverized coal at a by-product coke plant.

Design of Skirt-boards.—It is not usually practicable to let skirt-boards diverge or spread apart with the run of the belt because the lower edges would have to be trimmed to a curve to suit the contour of the belt on the

troughing idlers. There might be some gain in flaring the skirt-boards because the friction between them and the material would then be reduced; but the practical gain would be small and there would be the risk that accidental variations in the contour of the belt due to belt tension or the load of material would cause rubbing against the lower edges of the boards. It is better to keep the boards parallel and in the design of the loading arrangements, provide a support that will definitely fix the position of the skirt-boards with reference to the belt. If the design of the skirt-boards and their supports is left to chance or to a millwright new to such work, the job is not likely to be done right.

Feeding as Related to Belt Capacity.—The ordinary formulas for belt capacity (see p. 143) allow for some irregularity in feed, and even for some bare places on the belt occasionally; but for the best results the feed should be under control so that the load cross-section is uniform from end to end. With that arranged, the speed of the belt can be reduced below what the ordinary tables give and still the conveyor will deliver its rated load, or if necessary the belt can be run at a standard speed and convey more material

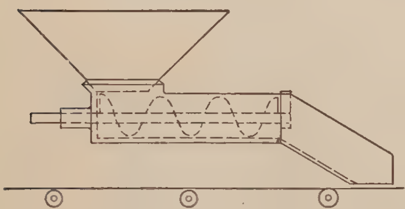


Fig. 127.—Screw Feeder.

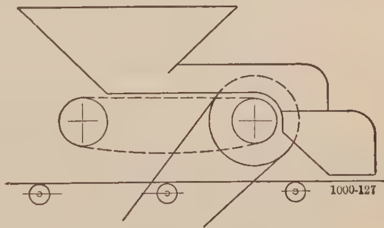


Fig. 128.—Apron Feeder.

than the standard rating. (For load cross-sections as related to width of belt, see Figs. 133, to 137.)

If the material is fine and lively a simple slide gate in a chute will control its flow, but this is not possible with lumps nor when lumps are mixed with fine stuff. For these some kind of feeder is necessary, especially when the material is drawn from bins or from track-hoppers. When crushers or pulverizers deliver to belt conveyors it is usually best to put the feeder over the crusher and then provide a hopper or a chute beneath it large enough to equalize any momentary rush of fine stuff through the crusher.

Belt Conveyor Feeders.—To pile an even load on the belt, the best feeder is one which has a steady forward delivery. The screw feeder (Fig. 127) will handle fine stuff, moist or dry, but it is not long-lived in gritty material and it is apt to be damaged if sticks, tools or large lumps accidentally get into it. The apron feeder (Fig. 128) gives a steady feed of any kind of material up to the largest lump of mined coal or ore. For heavy material it becomes costly. The reciprocating plate feeder (Fig. 129) is simple and cheap; it can be placed directly under a track hopper or rock dump without danger of damage from falling pieces or from mine

props, tools and such things which sometimes come in run-of-mine coal. It does not give a uniform load on the belt because it seldom makes over 30 strokes a minute, but if it delivers to the belt through a long chute or through a crusher, its intermittent or pulsating feed is made more nearly uniform. The shaker feeder (Fig. 130) is a modification of Fig. 129; its strokes are shorter and quicker and when it loads a belt through a short chute, the load cross-section is apt to be more uniform.

No mechanical feeder will pile a perfectly uniform load from end to end of a fast-moving belt. The rate of feet per minute or even per second may be uniform, but the variation of feed within a second may be enough to leave thin places or bare places on a belt traveling 10, 8 or even 4 feet per second.

The Stuart patent 1175190 of 1916 covers the idea of using two or more belt feeders in series to deliver to a high-speed conveyor belt. The first feeder receives from a hopper or other source of supply at low speed; the second feeder receives from the first, runs at much higher speed and delivers to the conveyor belt.

Other Loading Devices.—Several throwing devices have been suggested for delivering material to belt conveyors with a velocity that will lessen

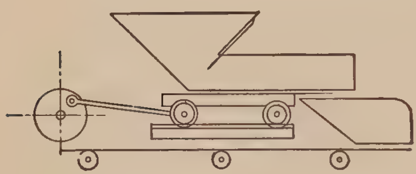


Fig. 129.—Reciprocating Plate Feeder.

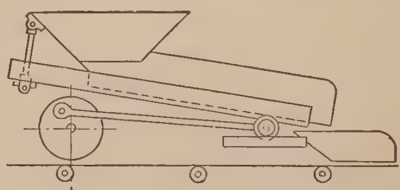


Fig. 130.—Shaking Feeder.

or avoid abrasion of the belt surface. In 1912 Thomas A. Edison patented a paddle-wheel feeder for use with fixed belt trippers where it was necessary to carry the material past the tripper. It was intended to remove the great objection to a series of fixed trippers, i.e., the wear on the belt from repeated reloading. The scheme was tried once on pulverized cement and discarded after a day's run. Similar methods are disclosed in the two Reinecke patents of 1911. Neither of them has come into practical use; they are costly and complicated and would take up more room than is necessary for a good chute. The cheapest and best way to speed material up to belt velocity is to let the force of gravity do it. There are, of course, places where chutes must be short, but it is well to remember that the expense of making a pit a foot deeper or an elevator a foot higher is often very much less than the continuing cost of the added wear on the belt.

Maxim.—It may be set down as a maxim that the proper duty of a conveyor belt is to convey, not to speed up, material.

Traveling Loading Hoppers.—When a belt draws its supply from more than one point, as from a number of openings under a long bin, or from a number of bins, there is a choice between using a number of chutes or a

single traveling hopper. In handling grain it is sometimes possible to use a number of fixed chutes, set with some clearance over the belt and with a pair of concentrators (Fig. 27) at each to prevent scatter and spill; but with coarser and more abrasive materials, fixed chutes with their skirt-boards would interfere with the flow of material past them and would be apt to damage the belt. The chutes can sometimes be arranged to swing clear of the belt when not in use, but it is frequently more satisfactory and often less expensive to use a traveling loading hopper.

Such hoppers for grain are comparatively light and can be pushed by hand. They consist of a box or funnel mounted on four wheels which travel on a track to which they can be clamped. The belt-loading chute is fixed at the proper distance above the belt and there are four concentrator pulleys carried on pivoted arms which by means of a hand-lever can be thrown into the operating position, or else thrown over to clear the fixed concentrators of the conveyor when the hopper is moved.

Traveling hoppers which draw coarse materials from bins are usually fitted with a feeder, because when the bin gate is opened wide it is often impossible to control the flow by hand so as to deliver a uniform load to the belt. In some cases the traveling frame is propelled by a motor which also drives a feeder of some kind. Fig. 131 shows a different kind where the conveyor belt furnishes the power for the traveling hopper. The belt passes around two pulleys at the end of the frame as in a tripper, and from a shaft driven by paper friction wheels on one or the other of the pulley shafts, power is taken by chain drives to an axle under the frame and to the head of the apron conveyor. A jaw clutch on the latter allows the feeder to be thrown out of gear during the traveling motion (Robb patent, 1904).

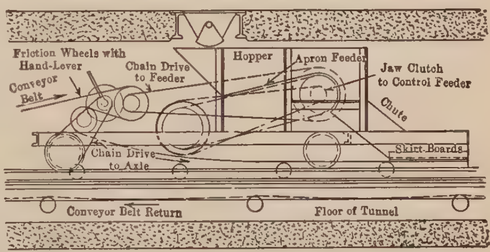


Fig. 131.—Traveling Hopper with Feeder Taking Power from the Conveyor Belt.

Loading Inclined Belt Conveyors.—The slope of an inclined conveyor is usually limited by the tendency of the material to roll downhill; hence, screened or sized material cannot be carried on angles as steep as where the lumps in a mixture rest on a bed of fines. An intermittent feed is objectionable when the angle approaches the maximum; single lumps fed to the belt may not be picked up promptly, but may tumble around between the skirt-boards for a time until a flat place happens to rest on the belt. The tail end of an intermittent feed may, for lack of a backing, become detached on the incline and dance up and down on the belt until the feed is resumed, or perhaps it may roll off. Conditions like these have fixed the angles at which it is practicable to convey various materials. For example:

Washed and screened pebbles	12°
Cushion-shaped briquettes	12°
Egg-shaped briquettes, less than	12°
Clean anthracite coal in domestic sizes	17°
Crushed screened coke	17°
Run-of-mine coal, crushed stone, run-of-oven coke	18°
Small crushed coal	20°
Bituminous slack when moist	22°
Tempered foundry sand	24°
Wet sand	27°
Fresh wood chips	27°

In general, these angles are 10° or 15° less than the slopes at which the material will rest on the belt without moving. They may be determined by experiment with some of the material and a piece of belt, but it is well to remember that when the angle of incline approaches the maximum for any material, there is danger that the material may at times slip. This may be due to irregular loading leaving bare places on the belt, to difference in the amount of moisture in the material, or to the depth of load carried.

The condition of the belt surface also has an effect on the way it picks up the load; if it is covered with fine dry dust, or if it is a new belt with the sulphur " bloom " still on it, or if on a cold day it is covered with minute frost crystals, the material may slip on it and refuse to go up the incline. Cleaning off the dust or getting rid of the frost by sprinkling salt on the belt are simple remedies which sometimes cure the trouble, but sometimes the angle is so steep that there is nothing to do but roughen the belt surface by fastening strips of belt on it or driving clinch rivets through it. As a general rule,—" better be safe than sorry "—keep the angle of incline less than the maximum.

In handling very gritty substances it is better to keep the angle well under the maximum at which it is possible to carry them without roll or visible slip, because in passing over the idlers there is an invisible slip or rearrangement of load which at steep angles scours the face of the belt. There have been cases where short steep belts handling gritty ore failed very rapidly from destruction of the rubber cover due to this invisible slip of material.

For economical speeds for inclined belts, see page **154**.

For loading chutes for inclined belts, see page **141**.

For skirt-boards for inclined belts, see page **137**.

For belt pulls for inclined belts, see page 113.

Choosing a Safe Angle of Incline.—In preliminary layouts of inclined belts it is well to assume an angle safely under the maximum, then there will be some leeway to increase the angle should it become necessary during the development of the details. Those experienced in such matters know that in the final drawings the foot of an inclined conveyor is generally lower and the head often higher than in the original plans. Both of these changes

increase the angle of incline unless the horizontal distance between terminals can be increased. It is often inconvenient or perhaps impossible to do that, since at slopes around 18° or 20° one foot of vertical height corresponds to three feet of horizontal distance and to change the height one foot without altering the angle means that the terminals must be separated horizontally three feet or more.

Much of the trouble with inclined belts comes, not from choosing the wrong angle for the material carried, but from the fact that in the development of the plans, the angle becomes steeper than was at first intended.

Change in Practice.—In the early days of belt conveying some engineers used angles of slope up to 6 in 12 ($26\frac{1}{2}$°) in order to use short conveyors or avoid the use of elevators. Sometimes the conditions of feed and of the material, crushed coal, crushed rock, etc., etc., were such that the conveyor worked satisfactorily, but more often there were difficulties and disappointments. In recent times practice has changed and angles over 20° are now rare.

Special Belts for Inclined Conveyors.—Robins in 1906 patented (810510) a rubber belt " provided upon its carrying surface with marginal and transverse elevations of the same height." The elevations, acting as rubber cleats, prevented the material from slipping on the belt and made it possible to run the conveyor at an angle steeper than ordinary. The marginal elevation allowed the belt to run over a snub pulley or the return idlers without bumping, but the whole construction was costly and it was not used more than once or twice. Fine material would lodge in the corners on the carrying surface of such a belt, and since a brush could not clean it well, there would be more drip on the return run than with a smooth belt.

Norton in 1907 patented a tailings stacker belt (see p. 19) in which the cleats or ribs are thick in the middle of the belt and do not extend to the edges of the belt.

Capacity of Belts as Affected by Troughing.—In 1896 Robins published a table of capacities of belts troughed at 45° in which cubic feet of material per hour at 100 feet per minute belt speed $=1.3W^2$ where $W=$ width of belt in inches. Fig. 132 shows to scale what the effective load cross-section, according to that table, looks like on the belt. It is probable that the rating was based on some actual records but tempered by caution lest inexperienced operators might expect too much from their belts.

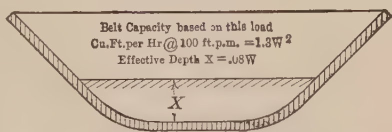


Fig. 132.—Load Cross-section According to Capacity Rating, 1896.

In modern practice capacities are rated higher. One formula used by Robins and others for belts on three-pulley or five-pulley idlers is cubic feet per hour at 100 feet per minute $=3.2W^2$; Jeffrey uses $3.5W^2$; Stephens-Adamson Co., $3W^2$; Link-Belt Co. uses a graduated formula in which the factor varies from 2.8 for 12-inch belt to 3.3 for 54-inch belt. These capacities can be expressed as square inches of load cross-section carried

on the belt; they are shown in Figs. 133, 134, 135, 136 drawn to scale. In every case the effective cross-sections represent much less than what might be piled at a slope of 4 in 10 (about 22°) with the lower edges of the

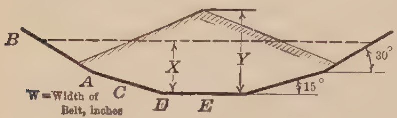


FIG. 133.

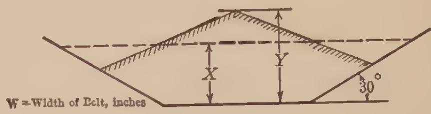


FIG. 134.

FIG. 133.

Capacity of belt on 5-pulley idlers based on $3.2W^2$ (Robins).
Depth of pile Y sloped 4 in 10 $=.205W$.
Depth X of equivalent flat load $=.129W$.
For 3-pulley idlers troughed 30°, $Y=.19W$; $X=.135W$.
For explanation of $ABCDE$, see page 82.

FIG. 134.

Capacity of belt on 3-pulley idlers, based on $3.5W^2$ (Jeffrey).
Depth of pile Y sloped 4 in $10=.216W$.
Depth X of equivalent flat load $=.144W$.

pile a few inches from the edge of the belt. The slope of 4 in 10 in these figures is assumed as representing an angle less than the angle of repose of most materials when stationary, and as an approximation to the shape of the average pile with the belt in motion. Shown as loads with a flat top,

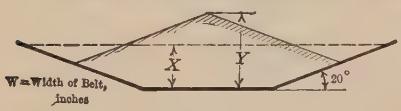


FIG. 135.

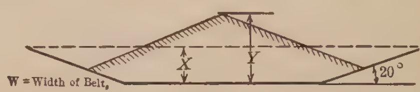


FIG. 136.

FIG. 135.

Capacity of belt on 3-pulley idlers, based on $3W^2$ (Stephens-Adamson).
Depth of pile Y sloped 4 in 10 $=.19W$.
Depth X of equivalent flat load $=.11W$.

FIG. 136.

Capacity of belt on 3-pulley idlers with wide center-pulley based on $3W^2$ (Main Belting Co.).
Depth of pile Y sloped 4 in 10 $=.18W$.
Depth X of equivalent flat load $=.094W$.

the level of the top would in every case (except Fig. 136) come below the line joining the edges of the belt in the troughed position. On the five-pulley idler the greatest possible load piled at 4 in 10 is about double the rated load (see Fig. 137). It might be attained if the belt speed were low, the conveyor level, and if the material, free from large lumps, were carefully fed to the belt through a wide chute. This combination of conditions seldom occurs; the supply of material to the feed chute or to the automatic feeder is seldom exactly uniform; the speed of the belt in relation to the feed is

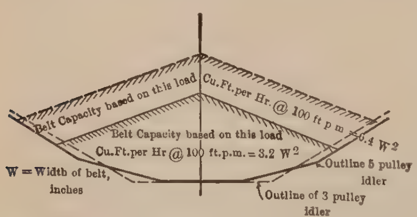


FIG. 137.—Comparison between Normal Load and Maximum Load on 3-pulley or 5-pulley Idlers.

often such that the load on the belt is not even and some parts may receive no load at all. Belts cannot be fed through a chute wider than $\frac{2}{3}W$ for fear of material scattering sideways or lumps rolling off the belt even where skirt-boards are used at the loading point. Again, on inclined conveyors, the agitation of the load in passing over idlers causes the peak of the pile to sink, and the pile assumes an angle of slope flatter than the one it assumes on a level run. The ratings adopted by manufacturers of conveying machinery and makers of belts are suited to average practice for belts horizontal or at angles of 20° or less; they cover the ordinary contingencies met with in operating conveyors. In some cases where the loading cannot be controlled with uniformity, or where fine lively material like cement is carried on an inclined conveyor, it is wise to expect capacities even lower than those given by standard formulas. One corporation that operated a coke and gas plant in the West used the following rule: "To find the capacity of a conveyor belt, make a diagram of the troughing idler, measure 2 inches down from the edges of the belt, draw a straight line across, calculate the area between it and the belt and from it the number of tons according to the material handled and the belt speed. *Then discount this one-third* and the result will approach the average maximum that one will get in practice." For a standard five-pulley idler, this rule comes to cubic feet per hour at 100 feet per minute $=2W^2$. It was based on long experience in handling run-of-mine coal, crushed coal and coke on many belts, some level, but mostly inclined at angles between 15° and 20°. It is a rule that is not recommended for general practice, but it is quoted to emphasize the fact that there are conditions where the ordinary safe rule cubic feet per hour at 100 feet per minute $=3.2W^2$ gives results not attainable in practice.

Capacity as Affected by Operating Conditions.—The factors which lower the capacity of a belt conveyor are generally external to the belt, such as delay in placing coal cars over the dump, delay in removing empty cars, trouble in getting coal out of the cars, trouble with crushers, delay in changing position of trippers, screens clogging, chutes filling up, stopping for lubrication or machinery troubles. All these reduce the time of actual conveying; experienced engineers always make some allowance for them and choose a belt wide enough to give the required capacity during the net working time.

The factors mentioned above are often more important in fixing the capacity of a conveyor than the amount of troughing the belt receives. It can be shown that for a given angle representing the way a certain material piles on the belt there is an angle of setting the troughing idlers which for a belt loaded to its edges gives the maximum cross-section, and hence the greatest belt capacity. For crushed coal, this angle is not far from 30° on three-pulley idlers, but whether the idlers are set at 20° or 35°, these full cross-sections are not very different from the theoretical maximum at 30°, and, as may be seen from the figures, the ordinary safe ratings derived from practice are much lower on any idler than would be

represented by the full cross-sections. Then, as has been said, the piling of the material on the belt varies with the nature of the material, the width of the feed chute and the angle of incline; taking all these things into consideration, it is not worth while to make rules which base carrying capacity on small variations in the angle of troughing. The ratings which are published by various engineering firms and which are shown in Figs. 133 to 136 inclusive, are in all cases lower than capacities at the fullest loading of the belts. The margins of safety are liberal and will allow some increase where the conditions are favorable for full loading and steady running. Unless the conditions are distinctly abnormal, it is wise to use the established rules and if it should appear in service that the material could be piled deeper on the belt, the belt speed can be reduced to what will give a good cross-section of load.

Table 26 gives carrying capacities of troughed belts based on cubic feet per hour at 100 feet per minute belt speed $=3.2W^2$.

Capacity of Belts Not Troughed.—If a belt of width W is run on flat idlers without troughing and is loaded to a distance of $.13W$ from each edge with material that piles at a slope of 4 in 10 (22°), it will hold just half the maximum capacity of a belt troughed as shown in Fig. 137. Since the ordinary safe rating of troughed belts is 50 per cent of the maximum possible loading (see Fig. 137) it is proper to rate flat belts by the same proportion of their maximum loading. Hence, capacity of flat belts in cubic feet per hour at 100 feet per minute $=1.6W^2$.

In Fig. 138 the outer line represents the maximum possible loading and

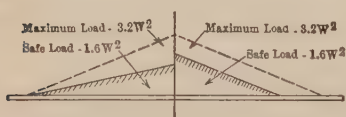


Fig. 138.—Comparison between Normal Load and Maximum Load on Flat Belt.

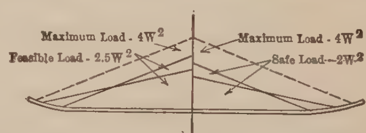


Fig. 139.—Comparison between Normal Load and Maximum Load, Belt on Flared Idlers.

the inner lines are the slopes of piles that hold half as much. These slopes are shown 12° (2 in 10) and 22° (4 in 10) and edges of the piles are $.15W$ or more away from the edge of the belt.

For capacities of flat belts of various widths, take one-half the values given in Table 26.

Capacity of Belts on Flared Idlers.—If a belt on Uniroll or commercial flared idlers (see Chapter IV) is loaded to within 1 inch of the edges of the belt with material that piles at 4 in 10, it will hold five-eighths as much as the maximum loading of a belt troughed as shown in Fig. 147. Keeping the same ratio between safe loading and maximum loading as for troughed belts, we get the rule: capacity of belts on Uniroll idlers, cubic feet per hour at 100 feet per minute $=2W^2$.

The right half of Fig. 139 shows by the outer line the maximum possible loading of such a belt on the basis stated above; the inner lines are the

TABLE 26.—CARRYING CAPACITY OF TROUGHED BELTS

Width of Belt, Inches	Cross-Section of Safe Load, Square Feet	Safe Carrying Capacity per Hour at Belt Speed = 100 Feet per Minute			Tons (2000 lbs.) of Material per Hour at 100 Feet per Minute Belt Speed							
					Weight of Material, Pounds per Cubic Foot							
		Cubic Feet	Cubic Yards	Bushels	30	40	50	60	75	100	125	150
12	.0768	460.8	17.07	370.3	6.91	9.22	11.52	13.82	17.28	23.04	28.80	34.56
14	.1045	627.2	23.23	504.1	9.41	12.54	15.68	18.82	23.52	31.36	39.20	47.04
16	.1365	819.2	30.34	658.4	12.29	16.38	20.48	24.58	30.72	40.96	51.20	61.44
18	.1728	1037	38.40	833.3	15.55	20.74	25.92	31.10	38.88	51.84	64.80	77.76
20	.2133	1280	47.41	1029	19.20	25.60	32.00	38.40	48.00	64.00	80.00	96.00
22	.2581	1549	57.36	1245	23.23	30.98	38.72	46.46	58.08	77.44	96.80	116.2
24	.3072	1843	68.27	1481	27.65	36.86	46.08	55.30	69.12	92.16	115.2	138.2
26	.3605	2163	80.12	1738	32.45	43.27	54.08	64.90	81.12	108.2	135.2	162.2
28	.4181	2509	92.92	2016	37.63	50.18	62.72	75.26	94.08	125.4	152.2	188.2
30	.4800	2880	106.7	2315	43.20	57.60	72.00	86.40	108.0	144.0	180.0	216.0
32	.5461	3277	121.4	2634	49.15	65.54	81.92	98.30	122.9	163.8	204.8	245.8
34	.6165	3699	137.0	2973	55.49	73.98	92.48	110.0	138.7	185.0	231.2	277.4
36	.6912	4147	153.6	3333	62.21	82.94	103.7	124.4	155.5	207.4	259.2	311.0
38	.7701	4621	171.1	3714	69.31	92.42	115.5	138.6	173.3	231.0	288.8	346.6
40	.8533	5120	189.6	4115	76.80	102.4	128.0	153.6	192.0	256.0	320.0	384.0
42	.9408	5645	209.1	4537	84.67	112.9	141.1	169.3	211.7	282.2	352.8	423.4
44	1.032	6195	229.5	4979	92.93	123.9	154.9	185.9	232.3	309.8	387.2	464.6
46	1.129	6771	250.8	5442	101.6	135.4	169.3	203.2	253.9	338.6	423.2	507.8
48	1.229	7373	273.1	5926	110.6	147.5	184.3	221.2	276.5	368.6	460.8	553.0

slopes of piles that hold one-half the maximum. These safe load cross-sections are shown with slopes of 12° and 22°; all of them, 12° or steeper, come well away from the edges of the belt.

If the conveyor is not inclined at more than 10° from the horizontal and if the loading is uniform it is possible to carry more than $2W^2$ over flared idlers. The left half of Fig. 139 shows safe load cross-sections one-fourth greater than those on the right half. For these conditions, cubic feet per hour at 100 feet per minute $=2.5W^2$.

Table 27 gives capacities of belts in cubic feet per hour at 100 feet per minute for $2W^2$, $2.5W^2$ and other ratings based on various belt loadings.

Capacity of Inclined Belts.—When the angle of the conveyor does not exceed 20° the safe carrying capacity of belts may be taken the same as for horizontal conveyors; but since the material on the incline tends to assume a flatter slope than on the level as seen in cross-section there is not in inclined conveyors the same margin between the maximum possible capacity and the capacity as given by the ordinary rule, cubic feet per hour at 100 feet per minute $=3.2W^2$.

For that reason and because it is harder to load inclined belts the capacity seldom if ever exceeds that given by the usual rule.

In estimating carrying capacities for inclined conveyors consideration should be given to the fact that belt speeds should sometimes be less than those for horizontal conveyors (see Table 30, p. 154).

Hourly Capacities.—For material like coal, which weighs about 50 pounds per cubic foot, the ordinary rule, cubic feet per hour at 100 feet per minute $=3.2W^2$, becomes: tons of coal per hour $=.08W^2$ at 100 feet per minute, or 8 per cent of the square of the width of the belt for every 100 feet of belt speed. Ton here means short ton of 2000 pounds.

Peak Load Capacities.—All rules for belt capacity apply to average conditions of loading where the feed is fairly uniform; but where a feeding device cannot be used or where the belt takes from a machine like a crusher whose rate of output for a period of some minutes may greatly exceed its average hourly rate, then the capacity of the belt should be considered on the *minute* basis and not the *hour* basis, unless the chute or hopper under the crusher is large enough to hold the excess of material during the time when the crusher output is above the hourly rate. (For an example based on this condition, see p. 116.)

An excess of belt capacity over the average hourly rate should also be provided where the conveyor receives material dug from cars or boats by a grab-bucket. When unloading begins from a full car or boat, digging is easy and the bucket will bring up more rapidly than later when the material gets low and the bucket cannot dig so well nor be handled so fast. In cases of this kind the output at first may greatly exceed the hourly average.

There are also conditions which, in order to maintain a certain daily total, require conveyors to handle per minute or per hour loads much greater than average. These are frequently overlooked in discussing the

TABLE 27.—CAPACITIES OF BELTS, CUBIC FEET PER HOUR, AT VARIOUS RATINGS—SPEED 100 FEET PER MINUTE

Inches	W^2	$1.2W^2$	$1.5W^2$	$1.75W^2$	$2W^2$	$2.25W^2$	$2.50W^2$	$2.75W^2$	$2.9W^2$	$3W^2$	$3.1W^2$	$3.2W^2$	$3.25W^2$	$3.5W^2$	$3.75W^2$	$4W^2$
12	144	172	216	252	288	324	360	396	418	432	446	460	468	504	540	576
14	196	235	294	343	392	441	490	539	569	588	608	627	637	686	735	784
16	256	307	384	448	512	576	640	704	743	768	794	819	832	896	960	1,024
18	324	388	486	567	648	729	810	891	940	972	1,004	1,035	1,053	1,134	1,215	1,296
20	400	480	600	700	800	900	1,000	1,100	1,160	1,200	1,240	1,280	1,300	1,400	1,500	1,600
22	484	581	726	847	968	1089	1,210	1,331	1,404	1,452	1,500	1,549	1,573	1,694	1,815	1,936
24	576	691	864	1008	1152	1296	1,440	1,584	1,670	1,728	1,786	1,843	1,872	2,016	2,160	2,304
26	676	811	1014	1183	1352	1521	1,690	1,859	1,960	2,028	2,096	2,163	2,197	2,366	2,535	2,704
28	784	940	1176	1372	1568	1764	1,960	2,156	2,274	2,352	2,430	2,509	2,548	2,744	2,940	3,136
30	900	1080	1350	1575	1800	2025	2,250	2,475	2,610	2,700	2,790	2,880	2,925	3,150	3,375	3,600
32	1024	1228	1536	1792	2048	2304	2,560	2,816	2,968	3,072	3,174	3,277	3,328	3,584	3,840	4,096
34	1156	1387	1734	2023	2312	2601	2,890	3,179	3,353	3,468	3,584	3,700	3,757	4,046	4,335	4,624
36	1296	1555	1944	2268	2592	2916	3,240	3,564	3,758	3,888	4,018	4,148	4,212	4,536	4,860	5,184
38	1444	1732	2166	2527	2888	3249	3,610	3,971	4,188	4,332	4,476	4,620	4,693	5,054	5,415	5,776
40	1600	1920	2400	2800	3200	3600	4,000	4,400	4,640	4,800	4,960	5,120	5,200	5,600	6,000	6,400
42	1764	2116	2646	3087	3528	3969	4,410	4,851	5,116	5,292	5,468	5,645	5,733	6,174	6,615	7,056
44	1936	2323	2904	3388	3872	4356	4,840	5,324	5,614	5,808	6,002	6,196	6,292	6,776	7,260	7,744
46	2116	2539	3174	3703	4232	4761	5,290	5,819	6,136	6,348	6,559	6,771	6,877	7,406	7,935	8,464
48	2304	2764	3456	4032	4608	5184	5,760	6,336	6,682	6,912	7,142	7,373	7,488	8,064	8,640	9,216
54	2916	3499	4374	5103	5832	6561	7,290	8,019	8,456	8,748	9,040	9,332	9,477	10,206	10,935	11,664
60	3600	4320	5400	6300	7200	8100	9,000	9,900	10,440	10,800	11,160	11,520	11,700	12,600	13,500	14,400
66	4096	4915	6144	7168	8192	9216	10,240	11,264	11,878	12,288	12,698	13,108	13,312	14,336	15,360	16,382

For any speed S, multiply $\times \frac{S}{100}$.

To change to bushels, deduct $\frac{1}{5}$.

To change to tons, multiply $\times \frac{\text{weight per cubic foot}}{2000}$.

capacity of the conveyor. The capacity should of course be equal to the "peak load" and not to the average rate.

Capacities of Grain Belts.—The last half of Fig. 140 shows, to scale, the load cross-sections corresponding to belt capacities as given by three authorities. Rule No. 3, bushels per hour at 100 feet per minute $=1.75W^2$, used by the Webster Manufacturing Co., represents what a belt with concentrators will carry under average conditions of loading from a single

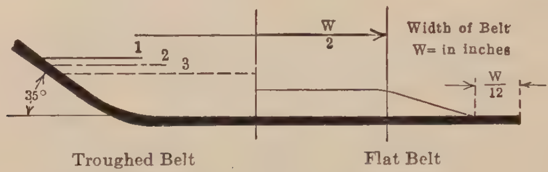


FIG. 140.—Capacities of Grain Belts According to Several Rules.

Rule 1.—Bushels per hour at 100 feet per minute $=2.5W^2$.
2.—Bushels per hour at 100 feet per minute $=2.2W^2$.
3.—Bushels per hour at 100 feet per minute $=1.75W^2$.

For flat belts where width of chute $=\frac{W}{2}$.

Bushels per hour at 100 ft. per min. $=1.2W^2$.

spout. When the concentrators come every 10 or 12 feet some additional load can be added by a second or a third spout without spill over the edges of the belt. There is no accepted rule for what may be added in this way, but the increase may be as much as 40 per cent without exceeding the loading shown by line 1, which represents the rule of H. W. Caldwell & Sons Co., i.e., bushels per hour at 100 feet per minute $=2.5W^2$. The load of grain carried by a fast-moving belt has a top which is nearly flat; the lines 1, 2, 3, therefore, show about what the loads look like according to the three rules.

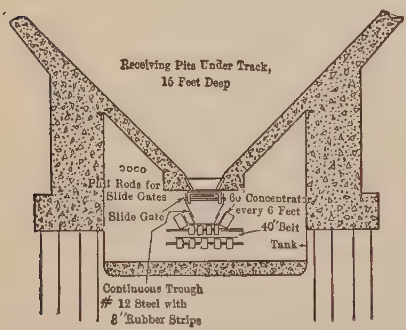


FIG. 141.—40-inch Belt with Continuous Loading Trough and 60° Concentrators, Public Grain Elevator, New Orleans (Ford, Bacon & Davis, Engrs.)

The angle of the concentrators (Fig. 27) has very little effect on the load cross-section in the diagram; steep concentrators with 60° angle are sometimes used where a belt is fed simultaneously from a number of spouts or from a continuous loading channel under a track pit (Fig. 141). Here the steep upturn of the edges of the belt prevents the grain from scattering sideways as it hits the belt. Usually the angle is 45° or less; 35° is common, but 30° and even 22° angles are in use. There is no doubt that the flatter the angle, the easier the action on the belt.

Grain belts narrower than 20 inches in 4-ply or even 3-ply thickness are relatively stiffer than wider belts in the usual 4-ply thickness, and show more tendency to flatten out between the concentrators. To get the rated capacities it is necessary to feed them with greater care and to keep a concentrator at each feed spout; and it is not usually possible to increase the rated capacity by feeding from a second or a third spout. Experienced grain-elevator engineers recognize these difficulties with narrow belts and do not use them at their published ratings, even the low ratings.

The right half of Fig. 140 shows the loading of a flat belt without concentrators, where the mouth of the spout is about half the width of the belt. With a clear margin on each side of $\frac{W}{12}$, the capacity of the belt in bushels per hour at 100 feet per minute $= 1.2W^2$.

Table 28 gives, in form for comparison, capacities of grain belts with concentrators as rated by four authorities. The ratings suggested by the author in the sixth column are lower than any of the others for the narrow belts.

TABLE 28.—CAPACITIES OF GRAIN BELTS WITH CONCENTRATORS

Belt Width, Inches	Bushels per Hour at 100 Feet per Minute					Advisable Speeds, Feet per Minute
	Caldwell, Line 1 in Figure	Goodrich, Line 2 in Figure	Webster, Line 3 in Figure	Weller	Safe Rule for Usual Conditions of Feed	
12	368	290	253	192	200	300
14	505	430	346	344	300	350
16	655	580	453	500	400	400
18	832	740	570	652	550	450
20	1025	910	706	800	700	500
22	1240	1090	866	960	860	600
24	1480	1280	1027	1096	1020	600
26	1745	1480	1200	1280	1200	600
28	2000	1680	1400	1456	1400	600
30	2320	1900	1600	1660	1600	650
32	2640	2140	1826	1860	1826	650
34	2960	2400	2053	....	2053	650
36	3360	2650	2333	2304	2333	700
38	3680	....	2560	....	2560	700
40	....	....	2800	2800	2800	700
42	4480	....	3120	3080	3120	750
44	4960	....	3400	....	3400	750
46	....	....	3790	....	3790	750
48	5920	....	4070	3960	4070	800

Speeds of Belt Conveyors.—The great advantage of belt conveyors over other forms of continuous conveyors is that they can run at higher speeds without increase of noise or much greater friction losses. So far as the transfer of material is concerned, all belt conveyors could, like grain conveyors, be run up to the speed at which the material would be blown off the belt by the resistance of the air. In practice, however, other factors

govern the speed. The width of the belt and its speed are, of course, determined primarily by the capacity required; but, in addition, the weight, size and nature of the material must be considered. If it is easily broken, like coke or anthracite coal or briquettes, the speed must not be so high as to discharge the material with violence at trippers or at end chutes. (For illustrations of end discharge, see p. 157.) If the material is not lumpy or abrasive, especially when it is free-flowing like grain or crushed coal, and the supply is regular, as from a large bin, the belt may be run fast. Then the loading chute can be arranged to deliver a uniform load to a fast-moving belt, but it is not economy to run a belt fast and have it only partly loaded as must happen if the material is lumpy; even though a feeder is used. A feeder of the apron or belt type (see p. 139) will deliver uniform quantities of material per minute, but in working on lumpy or mixed material, the fall of the material over the head of the feeder may vary from second to second, and on a belt moving 5 to 10 feet a second, it is often impossible to avoid a thin load or even bare spots at intervals on the belt.

Full Load Cross-section Desirable.—The aim should be to run the belt no faster than is necessary to give the capacity under the conditions of loading. A full cross-section of load at a slow speed means a deeper pile and proportionately less material in contact with the belt, hence less cutting and abrasion, less strain from pick-up of material (see p. 123) and less energy wasted in revolving shafting and gearing, foot shafts, snub shafts, tripper mechanism, etc. Fig. 142 shows a belt carrying only one-third of its normal rated load, and superimposed upon the light load, the other two-thirds of its normal load. Only a portion of the added load touches the belt. Actually the load is tripled with an increase of 60 per cent in the amount of belt surface in contact with the material. The full load distributes the load over more of the belt, the light load concentrates it in the middle; and when a belt is worn out in the middle, it is all worn out.

Fig. 142.—Comparison of Full Load Capacity and One-third Capacity.

The rated capacities of belts are, as may be seen from Figs. 133 to 137 inclusive, considerably less than what may be piled on the belt; in some cases where the feeding arrangements are favorable, it is possible to get a larger load cross-section than standard and hence a slower speed for the required capacity. After a belt conveyor has been in regular service for a short time it is always advisable to study the feed and the load, and then reduce the belt speed when it can safely be done.

High Speeds not Practicable for Narrow Belts.—Wide belts can be run faster than narrow belts because the wider the belt, the deeper the load and the less material proportionately comes in contact with the belt surface to cut or abrade it. On a narrow belt, a greater proportion of pieces touch the belt surfaces, hence, to lessen the injury to the belt, its speed should be less. Expressed mathematically, belt capacity for a given speed varies with W^2 while the belt surface passing the feed chute varies with W; the

ratio which measures the durability of the belt under the abrasion of material is, therefore, $\frac{W^2}{W} = W$. Reasoning from this, a fully loaded 48-inch belt at 900 feet per minute might suffer no more than a 16-inch belt at 300 feet per minute in handling the same kind of material; but since wide belts are often chosen because they will handle large lumpy material the speed for wider belts is held down to reduce the damage from cutting at the loading point.

TABLE 29.—MAXIMUM ADVISABLE SPEEDS FOR BELT CONVEYORS FOR COAL, ORE, GRAVEL, ETC.

Belt width, inches........	12	14	16	18	20	24	30	36	42	48
Belt speed, feet per minute.	300	300	300	350	350	400	450	500	550	600

Table 29 gives maximum speeds for belt conveyors according to the considerations mentioned above.

Speed Limited by Construction of Idlers.—The pulleys of commercial grease-lubricated idlers revolve about 60 to 75 r.p.m. for each 100 feet per minute of belt travel. The pulleys are not centered and balanced as transmission pulleys are, and at speeds over 400 or 500 feet per minute they are apt to be noisy and the idler stands shake loose from the vibration. For high speeds, the stands and pulleys of cast-iron grease-lubricated idlers must be better than the ordinary kind, in some of which the pulleys are not turned true.

Possible Maximum Speeds.—It is probable that in the future, belt conveyors for high capacity will be equipped with improved feeders and with ball-bearing or roller-bearing idlers that can run at high speeds. It may be possible then to run such belts almost up to the speed at which air resistance will blow the material off the belt.

Minimum Speeds.—As may be seen from Fig. 144, a speed of 140 feet per minute is sufficient to deliver material into a chute set just below and under the middle of the discharge pulley. Since tripper chutes and end-discharge chutes are generally set in that way, it is not necessary, so far as clean delivery of ordinary material is concerned, to run belts any faster than 140 feet per minute.

Jeffrey says, "Under 150 feet per minute speed, the cost of the belt conveyor per ton of bulk materials handled, even with a minimum ply of belt, commences to be uneconomical as compared with other types of conveyors equally suited to the operating conditions." Although this expresses a correct relation between belt conveyors and other types of conveyors as to first cost and operating cost, there may be reasons more important than costs which make a belt conveyor at 100 or 150 feet per minute preferable to any other conveyor at any speed.

Short Belts.—When belts are very short each foot of surface passes the loading point frequently, and hence the wear is proportionately greater

than in a long belt. For that reason a short belt should not be run faster than is actually necessary.

Speeds for Inclined Belts.—From what is said on page 134 about loading chutes it is clear that when belts are loaded on an incline the cutting and abrasion are likely to be greater unless the angle of the chute is flattened so as to deliver the material more nearly in the direction of belt travel. It is not usually possible to flatten the chute without risk of choking it, and, in addition, the flatter angle means less velocity of flow. Excessive wear on the belt from sharp or abrasive materials can be avoided by running the belt slower. In Fig. 143 if the chute is at angle of 40°, which represents a fair average, the best speed for material in it is V secant 40° when the velocity of the belt it feeds is V on the horizontal. If the belt is inclined at an angle A and runs at a speed S, then for the best delivery of material flowing from the chute, S should equal V secant 40° cosine (40°+A). Table 30 has been calculated on this basis; it shows that if for a certain material it is proper to run a horizontal conveyor at 400 feet per minute, to handle the same material on a belt inclined at 19° with no greater wear at the loading point, the speed should be no greater than 400×.67 = 268 feet.

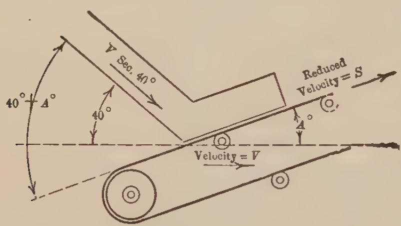


Fig. 143.—Reduction of Velocity of Inclined Belts to Lessen Cutting and Abrasion.

TABLE 30.—REDUCED BELT SPEEDS FOR LOADING ON INCLINES

Angle of incline, degrees	5	10	13	16	19	22
Percentage of normal speed for horizontal belt	91	83	78	73	67	61

A slower speed for steep belts also reduces the wear from slip on the incline (see p. 142), and since it lessens the jostling of the load over the idlers, it reduces the tendency of the material to roll or slip down the incline. (For a disastrous combination of 20° angle and 350-foot speed, see p. 117.)

Carrying capacities for inclined belts should be calculated with some reference to the lower speeds which are often advisable, especially in carrying stone, ore and other difficult materials.

Horizontal Loading Ends for Inclined Belts.—Conveyors inclined at about 20° for coke, large lump stone and other difficult materials have been built with a curve at the foot of the slope (see Fig. 111), so that the belt is loaded on a short run which is nearly or quite horizontal. This arrangement permits the belt to be run at normal speed, there is less cutting and abrasion of the belt surface and the material is not so likely to slip and roll while on the incline.

The curve should be so located with reference to the loading chute and be of

sufficient radius so that under no conditions can the belt lift off the idlers and be cut by the skirt boards.

The danger of lifting the belt is avoided if the path of the belt is changed by the use of pulleys as in Fig. 15, but the scheme has several drawbacks; it takes up room, it gives the belt several extra bends, one of them with the dirty side of the belt against the pulley, and there is the added cost of three or four deflector pulleys with their shafts, bearings and supports.

Speeds for Grain Belts.—The experiments by Westmacott and Lyster in 1865 showed that oats, bran and flour could be carried at speeds up to 480 feet per minute without being blown off the belt by the air resistance; they found also that wheat could be carried at 540 feet per minute safely, but that the chaff in it would be blown off. These trials were made with belts 12 and 18 inches wide, run flat except for concentrators at the loading points.

In modern practice wheat and corn are carried at speeds up to 800 feet per minute on belts 36 inches and wider; but on belts less than 20 inches wide the speed should be less than 500 feet per minute. The main reason is that in chutes suited to wide belts the flow of grain is rapid and regular and the belt can be loaded full, but in narrow chutes the flow is not so fast and even, and the narrow belt running faster than 500 feet per minute will not take a full load nor will the loading be uniform along the run of the belt.

The last column of Table 28 represents good practice in speeds of grain conveyors.

Speeds for Picking or Sorting Belts.—When belts are used for picking or sorting ores the speed must be low enough to permit the men alongside to pick out and turn over pieces on the belt without moving from their positions and without stretching too far. The speed is generally less than 50 feet per minute; for large lumpy material that must be turned over for examination the speed may be 25 feet per minute. (See also p. 193.)

CHAPTER VIII

DISCHARGING FROM THE BELT

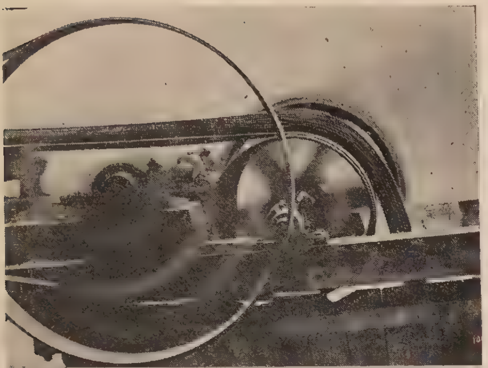
Discharge from Belts.—The simplest discharge from a belt conveyor is over the head pulley. The path of the material in leaving the belt is a parabola; its appearance at various speeds is shown in Figs. 144, 145, 146, 147. Rules and diagrams are given by manufacturers' catalogues for laying out the parabola by coordinates. Fig. 148 and Table 31 are Jeffrey's. The line $X-Y$ represents the approach of the belt to the discharge pulley. From the tangent point Y, where it meets the pulley, lay off distances L in inches equal to belt speed in feet per minute divided by 100. Distances A, B, C, etc., from the table measured vertically from the first, second, third point, etc., locate the curve.

Position of Chute.—In handling material easily broken, like sized anthracite coal, briquettes, screened coke, etc., it is desirable to place the chute so that it receives the discharge without shock and without excessive drop. At the same time, the top edge of the bottom plate of the chute should be set so that no pieces can jam in the gap between it and the belt, and in such a position that if the conveyor should be stopped while loaded, the dribble of material over the pulley in starting up at slow speed will not fall into the gap and be wasted or stick there and damage the belt. To a great extent these requirements are contradictory; for an easy delivery the chute should be at 1 (Fig. 148), but for safety to the belt it should not be placed higher than 2; if the material tends to stick to the belt and is not hurt by the drop, it is better to put the chute at 3. If there is a cleaning brush at the discharge pulley the chute must be further back, as in Figs. 174 and 175 so that at no time can material discharged over the head pulley fall directly on the brush.

Tripper Chutes.—In trippers where it is important to save height, the chute is usually placed at position 2, and the brush is placed below it so as to throw the fine stuff clinging to the belt back on the empty run as it leaves the tripper.

Choked Chutes.—When there is danger of material backing up in a choked discharge chute and then scraping the belt it is well to set the upper end of the chute some distance below the pulley, so that when the material does back up, it will overflow from the top of the chute before it reaches the belt on the pulley.

Dribble over End Pulley.—When a belt conveyor delivers to a bin through a tripper it is never safe to assume that all of the material will be discharged at the tripper and none over the end pulley of the conveyor.



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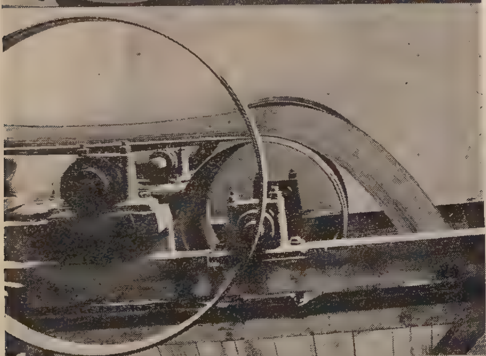

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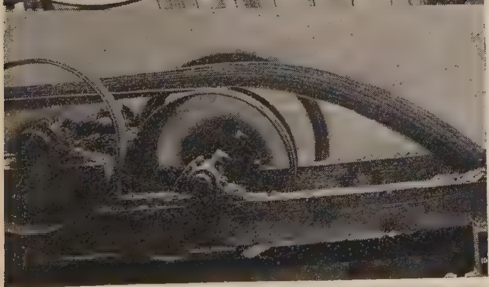

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Figs. 144–147.—Discharge at 24-inch Head Pulley of Belt Conveyor at Various Speeds.

144.—140 feet per minute.
145.—275 feet per minute.
146.—360 feet per minute.
147.—780 feet per minute.

The fine stuff which clings to the belt or which is brushed off in the tripper falls over the end pulley, and if the latter is not placed over the bin, the dribble outside of the bin may be a nuisance. If it is necessary to place the end pulley beyond the bin, an auxiliary chute may be provided to catch the dribble.

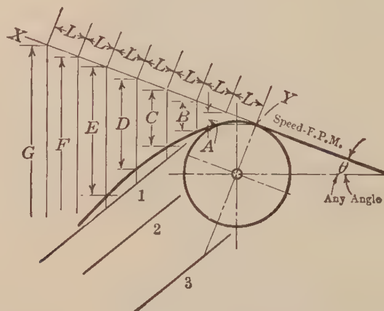


Fig. 148.—Path of Discharge over a Belt Pulley.

TABLE 31.—VERTICAL ORDINATES, INCHES, FOR USE WITH FIG. 148

A	.43	D	7.72	G	23.64	J	48.24
B	1.93	E	12.06	H	30.87	K	58.37
C	4.34	F	17.37	I	39.07	L	69.47

Plows or Scrapers.—When belts handle material like wood chips, fine coal, dry chemicals, etc., which are not abrasive and which can be carried with little or no troughing, it is possible to discharge them at various places along the run by plows or scrapers set diagonally across the belt (Fig. 149). The lower edge of the scraper should be a few inches above the belt and the actual scraping of material done by a strip of belting fastened to the board and touching the conveyor belt. Scrapers are sometimes hinged at one end and bear against a stop at the other, in which case they may be worked by a pair of pull ropes from a distant point; or they may be fitted between a pair of vertical guides as in the figure and transferred from place to place to change the discharge. With any construction, a chute or guard should be provided at each discharge point to prevent scattered material from collect-

Fig. 149.—Discharge from Flat Belt by a Diagonal Scraper.

ing under the upper run and fouling the idlers or else falling on the return belt.

This form of discharge works best with flat belts and light loads; it has been used with Uniroll idlers, but will not work with belts troughed more than 5° or 10°. It has been used in places where the first discharge from a belt came too close to the loading point to allow a tripper to be used, and on long conveyors outdoors where it was not convenient to install a traveling tripper.

Objections to Scrapers.—The general objection is, of course, the wear on the surface of the belt. A specific fault of the construction shown in Fig. 149 is that if the material is heavy, or the belt troughed, there is a component of pressure at the scraper which tends to force the belt out of line and off the idlers at the point of discharge. This can be avoided by making the scraper or plow symmetrical with a V point to discharge on both sides of the belt. With this construction, the discharge is not spread over so much length, but a disadvantage is that the ends of the rubber scraping strips which bear on the belt may catch at the belt splice or at worn places in the belt and do damage to the belt or else be worn off or torn loose.

Traveling Plows.—In some European boiler houses the bunker is served by a flat belt which runs through a movable carriage equipped with a V-point plow and a two-way chute. The movement of the carriage is controlled from the floor of the boiler room by a hand-power winch operating a pair of pull ropes. An indicator on the winch shows the position of the carriage, and it is not necessary for a man to go aloft to change the position of discharge. Besides that, the carriage is cheaper than a tripper and takes up less head room.

Comparison with Trippers.—As compared with a tripper, a plow causes more wear on the surface of the belt, but it does not tend to separate the plies of the belt by breaking down the friction layers between them by reverse bending, as a tripper with its small pulleys often does. If the material is fine and not abrasive, the life of the belt would ordinarily be determined by the life of its friction and not by wear on the surface. In such a case it may be ultimately more economical to use a scraper and put some wear on the belt surface so as to equalize in some degree the external and internal wear of the structure of the belt. Even though the life of the belt is shortened, it may still be economy to pay for belting rather than for a tripper and the necessary attendance, maintenance and repairs.

Distribution to Separate Bins.—When a belt is used to distribute material to a series of small bins a traveling tripper is at a disadvantage because too much time is lost in transferring the tripper from place to place and in making sure that the belt is empty when the tripper moves across the clear space between the bins, so that the tripper will not discharge material between the bins. There is also the expense of an attendant to set the tripper and make sure that it is clamped to the rails so tight that it will not work loose and travel with the belt. Fixed trippers set over each bin are sometimes used, but if the service is hard and continuous they hurt

the belt by the repeated reverse bending and by the repeated delivery of material back on the belt when the discharge point is beyond one or more of the trippers. The by-pass chutes furnished with trippers, traveling or fixed, do not deliver material back to the belt at belt speed and the wear on the belt is greater than from a well-designed loading chute (see Fig. 159).

When fine material is to be distributed to a series of small bins, as in delivering foundry sand to the hoppers of molding machines, a number of scrapers operated by pull-ropes will sometimes do the work at less cost and with greater convenience than any arrangement of trippers, traveling or fixed.

Other Discharge Devices.—An old device, patented by Palmer in 1888, is shown in Fig. 150. When the idler pulley is tilted, the material slides off the belt, but the discharge cannot be located with accuracy because it begins and ends as a dribble over the edge of the belt and the bulk of it is spread over some distance of travel. The scheme is used in a few places with belts less than 24 inches wide to discharge fine dry material into large bins where the position of discharge is not important. The return belt must be guarded against the spill from the upper run.

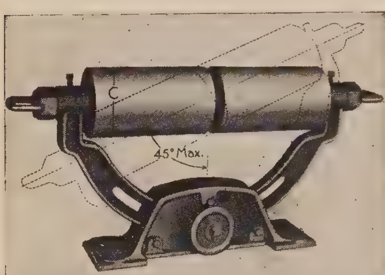


Fig. 150.—Tilting Idler to Discharge Material from a Belt. (Jeffrey.)

A device similar to a plow is shown in the Brotz patent of 1912. It consists of a horizontal disk of steel plate mounted on a vertical shaft and extending over the belt in such a way that as the disk revolves it will take material from the belt and deliver it to one side. It was tried once in a foundry to remove sand from a flat belt to a series of small bins. The disks were driven by frictional contact with the belt and could be moved into or out of operating position. They were not satisfactory, and were finally replaced by scrapers set diagonally across the belt and controlled by pull-cords from the floor below.

Discharge by Inverting the Belt.—Discharge at points along the run of a belt conveyor can be effected by the use of fixed trippers. If they are set in series to discharge at several points it is necessary to provide all but the last with a by-pass chute to reload material on the belt and fit them with flap gates to divert the stream, or else, as in Fig. 151, let the main chute fill up and overflow into the by-pass chute. This was the arrangement adopted in an American cement plant built about 1902 where a stock-house was filled by a belt running through a long series of fixed trippers. The travel of the cement in the by-pass chute was so short that it did not acquire any speed, and hence it wore the belt. In applying, in 1907, for a patent on a rotating-drum device to replace the by-pass chutes on these trippers and to throw the cement back on the belt with

some velocity in the direction of travel, the engineer of the plant says, " I find that the belts become very rapidly worn, since at each discharge station the material falling from the upper to the lower run of the belt requires to be moved from a state of rest to the speed of the belt." This has been the experience of others using fixed trippers with by-pass chutes, and at present they are seldom used. A traveling tripper bends the belt less than a series of fixed trippers, avoids all or most of the reloading on the belt and is likely to cost less than three or four fixed trippers with their chutes. At any tripper, the material must be lifted from 2 to 5 feet in order to effect a discharge. A traveling tripper does this once, but a series of fixed trippers does it oftener. Hence more power is required to drive the conveyor through such a series of fixed trippers.

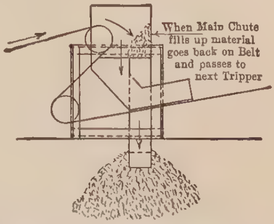


Fig. 151.—Fixed Tripper with By-pass Chute.

Use of Fixed Trippers.—When, however, a belt discharges to a series of small bins, or through hatches in a floor to tanks or chutes beneath the floor, it may not be convenient to use a traveling tripper, because in order to avoid spill on the floor the belt must be empty when the tripper is moved. In handling grain in American " elevators " it is customary to stop the flow of grain to the belt when the tripper is moved between spouts, but in some manufacturing plants it is impossible to shut off the feed to the belt without costly interruption of the processes. In such cases it is good practice to use a fixed tripper over each discharge hatch and suffer the added wear on the belt.

Stationary Trippers with Movable Pulleys.—Some of the objections to a stationary tripper with fixed pulleys can be avoided by making the upper

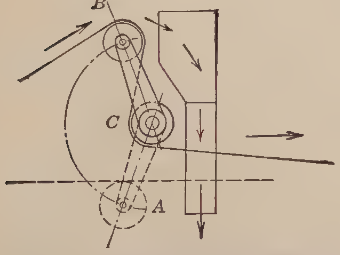


Fig. 152.—Stationary Tripper with One Pulley Movable to Effect Discharge or Allow Material to Pass By.

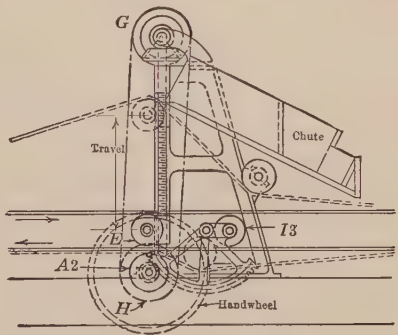


Fig. 153.—Cookman-Neall Tripper for Grain Belt.

pulley movable, either as in Fig. 152, where the pulley swings from its idle position at A to its operating position at B around an axis on which is

mounted the other pulley C, or as in the Cookman-Neall tripper of 1901. In this device, which was used at the Washington Avenue grain elevator in Philadelphia for a number of years, the power to lift the discharge pulley is taken from the return belt. Turning the hand wheel (Fig. 153) depresses $I3$ and forces the return belt into driving contact with $A2$. A chain running from H to G drives a pair of vertical screws through reversing clutches to raise or lower E.

Traveling Trippers.—The earliest trippers used in this country were stationary (see page 7). Movable trippers came into use after 1870. The first of these were similar to Westmacott's tripper (see p. 8). Fig. 154 shows one used at Duluth in the early eighties (T. W. Hugo, Transactions A. S. M. E., 1884). It consisted of a pair of cast-iron side frames that carried two pulleys which by means of the hand wheel and worm gear

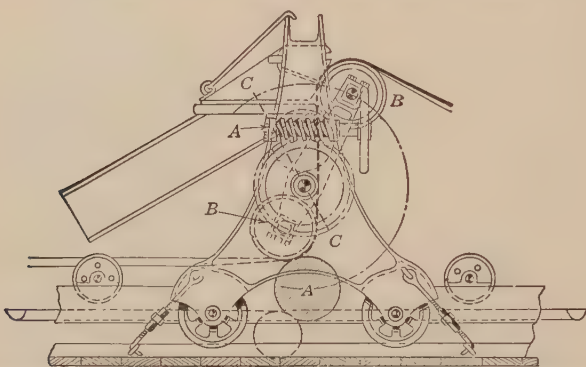


Fig. 154.—Hand-propelled Tripper with Pulleys Adjustable for Reversible Discharge.

could be rotated from the idle position AA to either of the working positions BB or CC. With the pulleys at AA, the loaded belt would pass through the tripper without being acted upon; with the pulleys at BB, grain coming from the right would be discharged into the chute; when the belt was reversed to carry grain from the left, the chute was transferred to the other side of the frame and the pulleys were swung to the position CC. The frame was pushed by hand to the discharge point, while the belt was slack, and was fastened to the floor by hooks and turnbuckles; the pulleys were raised to the working position and then the conveyor take-ups were tightened.

Self-propelled Trippers.—William B. Reaney (see p. 8) designed the first self-propelled tripper in 1876. It was like Fig. 154 in having the pulleys adjustable for discharging a reversible belt, but it was new in taking power from the conveyor belt to propel the frame backward and forward.[1]

[1] Communicated to the author by Mr. George M. Moulton of Chicago, who with his father, John T. Moulton, built an elevator at Duluth in 1869, the Canton elevator in 1876 and many other important grain elevators.

After a few years of successful use at the Canton Elevator, self-propelled traveling trippers came into general use, and for simplicity in construction they were generally made with pulleys on shafts fixed in position. In the older hand-propelled trippers it was necessary to swing the pulleys out of position in order to push the frame easily, but that was not necessary with power-driven trippers. At first, separate friction clutches or jaw clutches

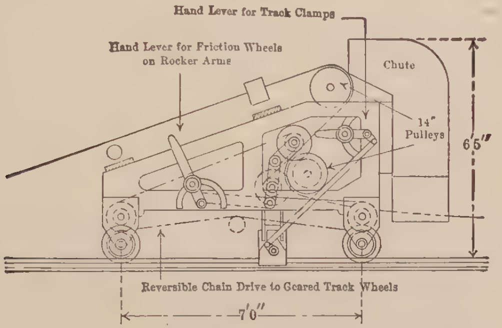


FIG. 155.—2-pulley Tripper for High-speed Grain Conveyor.

were used on each pulley shaft to drive chains leading to the track wheels, but later the modern style with paper and iron friction wheels came into use. Fig. 155 shows a modern friction-driven tripper for high-speed grain belts with geared track-wheels. When a tripper is required to discharge from a reversible belt it is simpler and more convenient to use four pulleys rather than the two movable pulleys of the older style. Fig. 156

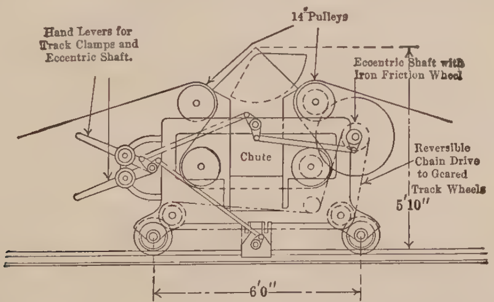


FIG. 156.—4-pulley Tripper for Reversible High-speed Grain Conveyor.

shows a modern four-pulley tripper controlled by two hand levers, one for the friction wheels, one for the rail clamp. The hood over the chute is pivoted and can be tilted to receive the grain from either side. This style of tripper is seldom used for material other than grain; the pulleys are only 12 or 14 inches in diameter, too small for belts thicker than the 4-ply generally used on grain conveyors. A tripper typical of those used for materials like coal, crushed stone, etc., is shown in Fig. 157. The frame is

low, the machinery simple, and since the opposite track wheels are mounted on separate pins instead of an axle the tripper tracks can be set low without interference between the tripper and the troughing idlers. The friction wheels of compressed fiber board at *A* and *B* are 7 inches in diameter; the operating shaft *C* is mounted in eccentric sleeve bearings which by means of the hand lever *D* can be moved in one direction or the other to force the iron friction wheel *E* into contact with *A* or *B* to propel the frame either backward or forward. There is a chain drive on each side of the frame connecting the operating shaft with one or both track wheels; the friction wheel *E* is 18 inches in diameter, and the chain drive is such that in a tripper

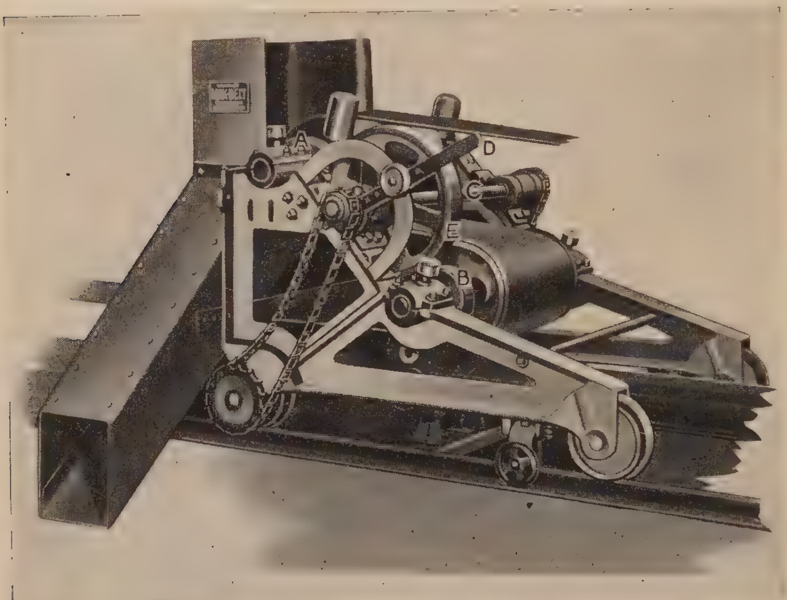


FIG. 157.—Self-propelled Tripper with Friction Drive. (Link-Belt Company.)

equipped with 16-inch belt pulleys, the forward travel is about 22 feet per minute and the backward travel 26 feet.

Worm-geared Trippers.—The original worm-geared tripper patented by Ticknor and Baldwin in 1904 has on one end of the lower pulley shaft a bevel gear which meshes with two bevel pinions on a shaft lying parallel to the belt and driving each truck axle through a set of worm gears. The bevel pinions on the worm shaft may be clutched to it by the motion of a sliding jaw so as to turn the worm shaft in either direction, or else they are shifted bodily along a key to engage, one or the other, with the driving gear. The driving mechanism is all enclosed and the appearance is good. The positive drive through gears is objectionable for high-speed belts, but when the belt speeds are under 350 feet per minute, and the tripper pulleys

are of good size, the arrangement works well without racking the tripper too much.

Automatic Self-reversing Trippers.—In the worm-geared tripper the travel of the clutch parts is very short and the motion is given by a short lever projecting vertically from the gear box. It is therefore easy to give the tripper an alternating backward and forward motion during the travel of the belt by arranging stops along the run of the conveyor to engage the shift lever. This makes an automatic self-reversing tripper that will distribute the discharge from a belt over the length of a bin or a stock pile without the attention of an operator to set and reset the tripper.

Similar automatic self-reversing trippers are made by the Stephens-Adamson Manufacturing Co. with paper and iron friction wheels, by the Link-Belt Company with shifting clutches, by Jeffrey Manufacturing Co. with bevel gears and a reversing shaft, etc.

In filling very long bins, an automatic self-reversing tripper is a convenience; but in boiler houses of ordinary size it is often better to avoid the complication of parts and use a simple tripper which can be clamped to the track while it fills a part of the bin. In many cases an automatic self-reversing tripper does not reduce the cost of attendance; it does add to the first cost of the plant and increases the cost of up-keep.

Tripper Pulleys.—Old grain belt trippers had 12-inch diameter pulleys as "standard." The custom persists, and some are still built so. They work fairly well with the 4-ply belts generally used for grain conveyors, but there is no doubt that the belts would last longer if the pulleys were 4 or 5 inches in diameter for each ply of the belt (see p. 127), that is, 16 or 20 inches in diameter for 4-ply belts. Grain-conveyor belts do not often fail by cutting or abrasion, but rather from splices pulling apart and from separation of the plies due to failure of the friction rubber. Both of these troubles are aggravated and the life of belts shortened by the use of small pulleys in trippers. Large pulleys cost a little more, but they save more than their added cost in the increased life of belts and the avoidance of belt troubles and repairs.

For materials other than grain, the case is different; the belts are generally thicker than 4-ply, and the splices hold better; moreover, the life of the belt is often determined by the surface wear rather than the separation of the plies. In such cases it would not always be economy to make pulleys 4 or 5 inches in diameter per ply of belt, especially since belts for heavy work are frequently 6-, 7- or 8-ply thick. The tripper would become too large and too heavy, and the life of the belt would not be economically prolonged. But while there may be doubt about making tripper pulleys 30 inches in diameter for 6-ply belts, for instance, it is certain that the ratio of diameter to ply should never be less than 3 to 1, that is, 18-inch diameter for a 6-ply belt. There are many cases where a ratio of 4 to 1 would be better, even in the class of work considered here. Some judgment is needed in selecting the sizes of tripper pulleys; manufacturers' "stock sizes" are frequently too small for good work.

Tripper chutes are made in various ways; for discharge to one side, to both sides by a split discharge, or alternately, or in such a way as to put the material back on the belt. Ordinary two-way chutes on trippers for coal, stone, ores, etc., are generally fitted with a flap gate (Fig. 158) for discharge to either side alternately. When the material is put back on the belt for discharge over the end pulley of the conveyor, two flap gates are used (Fig. 159). The delivery of material to the belt in this way is not according to the best methods of loading a belt, because the chute is necessarily short and the material does not acquire much velocity in the direction of belt travel.

Tripper chutes are often set with the upper edge at position 2 (Fig. 148) so as to save height in the frame and lift of material in passing through the tripper; there are, however, several disadvantages. If the belt is stopped while loaded, starting again at slow speed causes material to fall into the space between the belt and the chute, and if the pieces wedge there and are sharp and angular the belt may be cut or torn. When a chute is placed at position 2, the clearance is generally made small to catch all of the material from the belt. It has happened that a blistered belt has caught at the edge of the chute, pulled the tripper loose from the track where it was clamped, and caused a serious accident. It is better to put the upper edge of the chute at a position lower than 2 so as to catch all of the spill and yet with clearance enough to avoid catching torn or blistered places on the belt, or a loose belt-fastener.

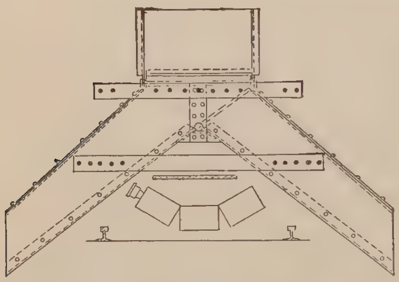


FIG. 158.—Tripper Chute to Discharge to Either Side.

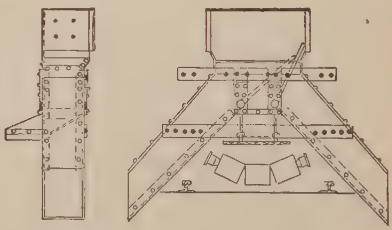


FIG. 159.—Tripper Chutes to Discharge to Either Side or Back on the Belt.

Tripper Brushes.—Trippers that handle sharp gritty material, especially if wet, should be fitted with rotary cleaning brushes, or else the lower pulley will force sharp particles into the cover of the belt. Since the brush must revolve *against* the travel of the belt, it is usually driven from the upper pulley shaft by a chain which can be shortened when the brush is raised, to compensate for wear.

Belts handling sharp wet sand have been ruined for lack of a brush in the tripper (see p. 199). It must be said, however, that a brush in a tripper needs constant care and attention to keep it in working order.

Belts Running Crooked through Tripper.—When a belt, instead of running straight through a tripper, runs off to one side so much as to scrape its edges, it is well to look to the gauge of the tripper rails. If there is too much clearance between the rails and the wheel flanges, the tripper frame may pull out of square with the belt. The belt will also run crooked if the ends of the belt are not cut square at the splice or if the tripper frame is not stiff enough to resist distortion.

Trippers should be made so wide that if the belt does run off the pulleys for an inch or two it will not scrape its edges on the side of the discharge chute or on the propelling mechanism of the tripper. Edge rolls are sometimes used to guide the belt in a tripper, but they should be avoided if possible, as they cause wear on the edges of the belt.

Trailers.—Large trippers for heavily loaded belts are sometimes made with trailers which are rear extensions of the tripper frame carrying several

FIG. 160.—Large Belt Tripper with Trailer Extension.

sets of idlers intended to support the belt as it rises into the tripper (Fig. 160). In many cases the trailer is of little or no use, because when the load on the conveyor changes, the belt lifts off the trailer as shown in the figure. The same thing may happen when the tripper approaches the head of the conveyor. The belt tension is greater there, and as a consequence the belt sags less on the lift into the tripper and does not touch the idlers on the trailer.

Unusual Forms of Traveling Trippers.—In 1901 a patent was issued to Humphreys on a tripper "provided with idler pulleys over which the belt runs, and means actuated by the one or more of said pulleys for giving travel to the tripper." This patent was granted without full knowledge of the prior use of traveling trippers by Reaney and others; it led to misunderstandings and stimulated the invention of several traveling trippers propelled by haulage ropes instead of taking power from the belt Some were made and used, but the driving machinery was more complicated than that of a self-propelled tripper; with a wider knowledge of what had already

been done in grain-belt trippers, the apparent need for these special trippers passed away.

Proal, in 1907, patented a worm-geared tripper for a reversing belt with an upper fixed pulley from which the frame was driven, and two other pulleys on a swinging frame to deflect the belt in one direction or the other. Morton, in 1907, patented a similar device with two pulleys rocking around an intermediate center. In 1907 Robins and Baldwin patented a tripper designed to travel back and forth over a series of separated bins and discharge to the bins only and not in the spaces between the bins. The chutes of the tripper were to be made large to act as storage reservoirs; gates at the lower ends of the chutes were closed while the tripper traveled between the bins, but open while over the bins. Neither of these three devices has come into practical use.

The Messiter patent 841558, of 1907, discloses an automatic self-reversing tripper designed to travel in one direction at the speed of the belt,

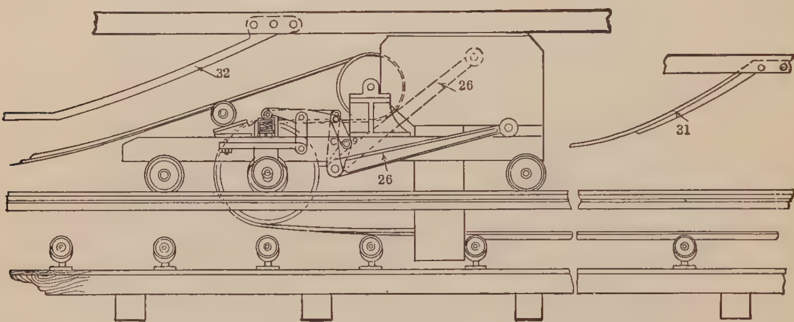


Fig. 161.—Tripper for High-speed Travel and Quick Reverse.

and at a definite speed in the opposite direction, with a prompt reversal at each end of the travel. In Fig. 161 the tripper frame carries four idle wheels which travel on a track somewhat higher than the belt idlers. The upper tripper pulley is fixed in position; the lower pulley is mounted on a shaft capable of a short distance of vertical movement. This movement is controlled by an arm 26 and a set of toggle levers. When the arm is up, the toggle is closed and the shaft is forced down, so that a flanged wheel at each end of the shaft engages the track rail, and since the tripper pulley which is fastened to the shaft travels counter-clockwise, the frame travels against the motion of the belt and discharges material. At the end of the travel a cam track 32 depresses the arm 26, opens the toggle joint and allows a pair of springs to lift the track wheels from the track against a pair of brake blocks. These brakes stop the rotation of the lower tripper pulley without shock, and the tripper reverses and travels forward with the belt at belt speed. On the forward travel, no material is discharged from the belt, but at the end of this travel the arm 26 is raised by the cam track 31, the brakes are released, the track wheels are depressed, the tripper is reversed without shock and the discharge of material begins again.

This tripper is used in the Messiter patent bedding system at a number of copper smelters to form longitudinal piles of ore, fuel and flux in layers. With a definite and uniform load on the belt, a positive rate of tripper travel and a prompt reversal at each end of the travel, the various materials are deposited uniformly so that each foot of length of the pile is the same in composition, and when properly reclaimed can be treated alike in the smelting operations. This tripper is more positive in its travel than one driven by friction wheels, and will reverse at high speed without shock, a feature impossible with trippers driven by clutches and gears.

Length Required for a Tripper.—Diagrams of traveling belt trippers given in manufacturers' catalogues show the distance from the chute at the front end to the point at which the straight slope of the belt rising into the tripper meets the line of the horizontal belt. This distance (dimension *A*, Fig. 162) is from 14 to 18 feet; but in a horizontal conveyor, the distance *B* from the center of the chute to the point at which the belt begins to lift

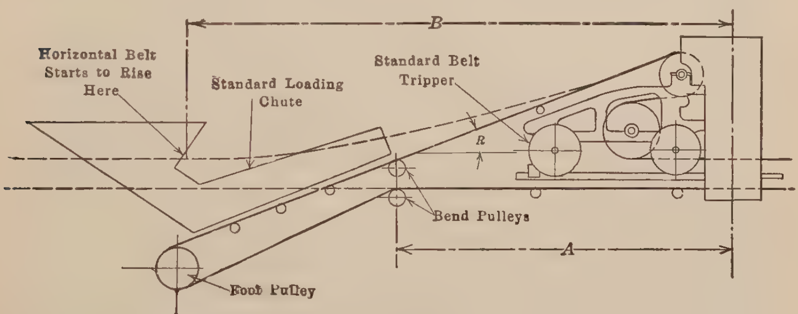


Fig. 162.—Location of First Point of Discharge through a Tripper with Reference to the Loading Point of a Belt.

off the idlers and curve up into the tripper may be 5 to 12 feet more than the distances given in the tables and diagrams. This is shown in Fig. 160 where the length is over 25 feet.

Location of First Discharge Point.—The point at which the belt begins to lift off the idlers in a horizontal conveyor helps to fix the place of the first discharge through a tripper, because the tripper can not come any nearer to the loading point without danger of lifting the belt under the loading chute or the skirt-boards. If the belt should be lifted by the tripper's coming too close, it will be scraped or cut and perhaps ruined in a few minutes.

It is possible to prevent the tripper from coming too close to the loading chute by using stops clamped to the tripper rails, but there is always some uncertainty about the point at which the belt will begin to lift, because if it is lightly loaded, or if it is pulled tight by careless use of the conveyor take-ups, or in an effort to get more driving effort at the drive pulley, the belt will lift sooner and span over a greater distance.

To avoid these difficulties, it is customary to depress the foot ends of horizontal conveyors when the first discharge comes close to the loading point. In Fig. 162 *B* represents the distance at which the belt begins to lift from a horizontal run, but by humping the conveyor and loading on a short incline the distance is reduced to *A*, and the point of first discharge is brought closer to the loading point by approximately *B—A*. Stops should still be used to limit the movement of the tripper (see Fig. 163) because while the depressed end of the conveyor does prevent fluctuations in belt tension from affecting the angle of approach to the tripper, it can

FIG. 163.—Belt Conveyor with Depressed Loading End, and Stops to Limit Travel of Tripper.

not be depended upon to stop the travel of the tripper and prevent it from lifting the belt off the idlers on the short incline.

Horizontal conveyors with depressed loading ends are often used in filling short bins where the head room under a roof is small or where it is important to bring the position of first discharge as close as possible to the loading point. There are, however, some drawbacks to this plan; it introduces another bend in the belt, and, what is sometimes more objectionable, loading on an incline. When the short end is inclined at 17° or 18° to match the angle of straight approach to the tripper the material cannot be delivered with proper velocity in the direction of belt travel. At high speeds of belt travel the lumps roll around longer before they acquire belt speed and the skirt-boards must be made longer on that account. All this means added wear on the belt.

When the position of first discharge is more than 25 or 30 feet from the end of the skirt-boards in a horizontal conveyor it is not generally necessary to depress the loading end, but for shorter distances it is better to depress

the end and suffer the added wear on the belt rather than run the risk of spoiling the belt by accidentally lifting it under the loading chute or the skirt-boards.

Fig. 164 shows the loading end of a 30-inch belt conveyor in a power house where the foot pulley could not be depressed below the level of the

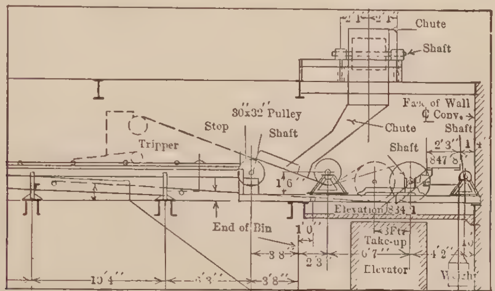


FIG. 164.—Depressing the Upper Run of a Belt to Permit Loading Close to Travel of Tripper. (Heyl and Patterson, Inc.)

conveyor. This arrangement with one depressor pulley accomplishes all that a depressed end does and puts no more bends in the belt. The first discharge in this case comes 14 feet from the end of the bin and 23 feet from the foot pulley.

Devices to Increase the Range of Discharge from a Tripper.—In storing soft coal, where it is desirable to limit the depth of pile to reduce the risk of fire, or where it is desired to fill a wide storage building from a central conveyor, or where there are objections to putting the conveyor on a high frame or trestle, it is possible to increase the quantity stored by fitting the tripper with an auxiliary conveyor to carry the discharge off to one or both sides. Fig. 165 shows the Blaisdell device patented 1903; the tripper is mounted on a wide-gauge track and fitted with an inclined belt conveyor to pile the material off to one side. In the Moss patent of 1907 the scheme is similar, but the boom which carries the conveyor is mounted on a turntable in the tripper frame and is hinged to move up and down. The auxiliary belt will therefore pile material over a considerable area at one setting of the tripper. In another arrangement used by Weller Manufacturing Co., Stephens-Adamson Manufacturing Co. and others, the tripper carries a reversible belt conveyor on a frame which can be racked in and out at right angles to the main conveyor, and thus discharge material at various distances on each side of the center line.

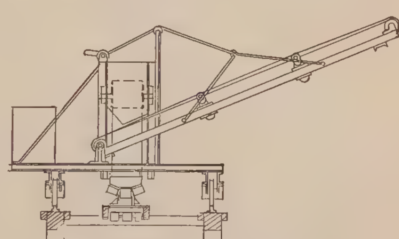


FIG. 165.—Traveling Auxiliary Belt to Increase the Range of Discharge from a Conveyor.

Extensions of these ideas are shown in other arrangements for distributing materials. The Blaisdell patent of 1902 discloses a main conveyor running along a row of leaching or cyaniding tanks and discharging to a shorter conveyor which spans the width of the tank. This conveyor may carry a tripper for distribution to the tanks and its traveling frame is connected to the tripper of the main conveyor so as to maintain the two conveyors in proper relation to deliver to any point in any of the tanks. In the Dodge device of 1904 a belt conveyor on a low structure discharges to a flight conveyor on a traveling cantilever frame which also carries the tripper pulleys for the belt. When the flight conveyor is inclined at the angle at which the material naturally piles, a high pile can be formed, the volume depending on the length of the belt conveyor and the length of the flight conveyor.

Stuart Devices.—Several patents granted to F. L. Stuart since 1916 show means to take material from a belt conveyor at ground level and form a storage pile alongside it. Fig. 166 (patent 1331464 of 1920) shows

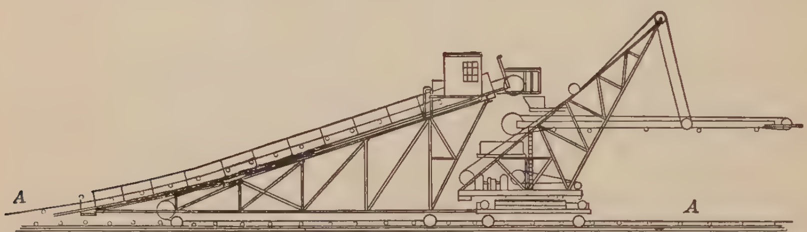


Fig. 166.—Forming a Pile Alongside a Belt Conveyor by Means of a Suspended Belt Taking the Discharge from the Main Conveyor.

one scheme. The conveyor A runs through a wheeled frame which is in effect a high tripper. A revolving tower mounted on power-driven trucks moves the tripper frame, and contains a pivoted boom which carries a belt conveyor that receives material from A. Several machines of this design have been built by the International Conveyor Corporation.

An adaptation of this device is used by the Baltimore and Ohio Railroad at Locust Point, Baltimore, for loading cars. The tripper frame is lower and is fitted with a double-jointed arm, each section of which carries a short belt conveyor capable of 270° angular movement. In a manner similar to that of the Manierre loader (Fig. 191), the conveyor arm can be inserted through the doorway of a box car and the material piled at either end of the car.

The coal-shipping pier of the Baltimore and Ohio Railroad at Curtis Bay, Baltimore, has four 60-inch 12-ply rubber belt conveyors about 1000-foot centers (see Fig. 101) which discharge coal at the head of long inclined loops, which are practically trippers, to reversible shuttle belts carried by traveling towers that load the ships, Fig. 167. The shuttle belts can be racked in or out and raised or lowered to suit the height of

ship's freeboard and the conditions of loading. The inclined tripper loop is connected to the frame of the shuttle belt and is hinged at its lower end at wharf level, so that the height of the tripper is varied to suit the elevation of the shuttle belt. This and other features of the equipment of this pier are covered by Stuart's patents 1192016 of 1916 and 1241053 of 1917.

At the coal pier of the Gulf, Florida and Alabama Railroad at Pensacola, Florida, a 42-inch belt carrying 600 tons run-of-mine coal per hour at 475

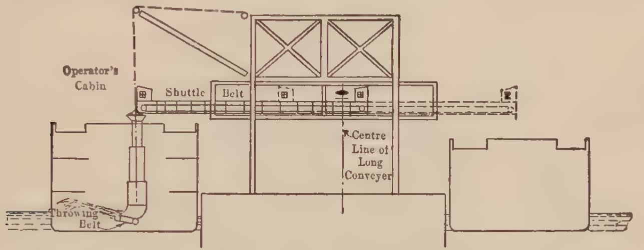


Fig. 167.—Loading Ships by Means of Shuttle Belts Serving Longitudinal Conveyors on the Wharf.

feet per minute loads ships through a chain and bucket elevator 75 feet high that discharges into adjustable chutes. The elevator is mounted in a power-driven traveling tower which, in addition, carries the loading chutes and a pair of pulleys which act as a tripper. As the tower travels on the pier, the coal on the belt is discharged into the foot of the elevator.

Shuttle Conveyors.—In some places where the discharge is spread over a comparatively short distance, or where it is not convenient to use a tripper, it is possible to vary the point of discharge by using a belt conveyor in a portable frame, so that the conveyor receives material at various points along its length, but always discharges over the end pulley. From the fact that they move back and forth, these machines are called shuttle conveyors. In the simple form shown in Fig. 168 the shuttle belt is mounted in a short frame pushed by hand so that the discharge over the end pulley will fall into bins No. 1, No. 2 or No. 3, while a by-pass chute from the

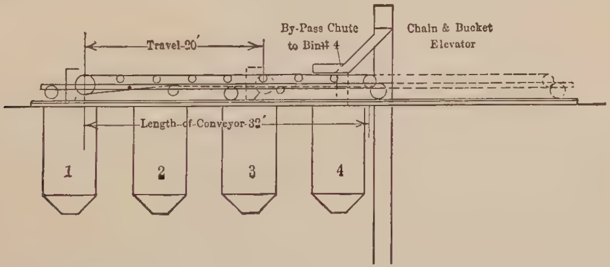


Fig. 168.—Simple Form of Shuttle Belt Conveyor Serving Three Pins.

head of the elevator discharges into No. 4. This device does what might otherwise be done with a fixed conveyor with a depressed loading end and a traveling tripper or a series of fixed trippers; in some places it is better than any of these arrangements.

Where the material is to be delivered on each side of the feed point, the belt is made reversible in direction, and it is loaded through a two-way chute with a flap gate in it. In this way it is possible for a shuttle conveyor of length L to discharge over a length of pile or bin nearly equal to $2L$.

The original shuttle conveyor patented by Bartlett and Overstrom in 1899 was driven and propelled by a rope drive; as now made, the conveyor

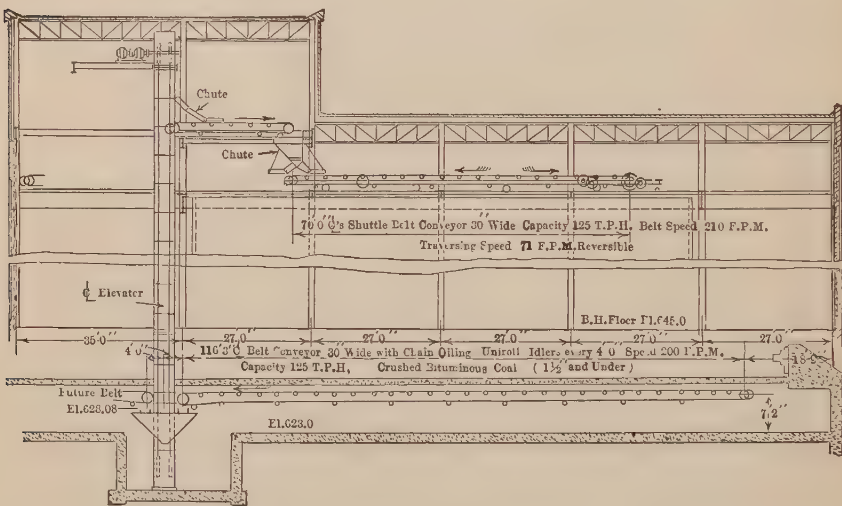


Fig. 169.—Long Boiler House Served by Shuttle Conveyor. (Link-Belt Company.)

is generally driven by electric motor. When the frame is short, or seldom changed in position, the machine is moved by hand, but large frames are more conveniently propelled by a separate motor or by a rope pulled by a winch-head mounted on the machine.

In Fig. 167 the shuttle frame is racked in and out by power. The reversible belt conveyor in it receives from another conveyor running the length of the pier, and it discharges material to ships lying on either side of the pier. All the motions of frame and conveyor are controlled by an operator in the cabin on the outboard end of the frame.

Fig. 169 shows a boiler house served by an elevator which stands at what will be the middle of the house when the other half is built. A shuttle conveyor 70 feet long serves the present bin which is 104 feet long. When the bin is doubled in length, a short conveyor similar to the present one will run to the left of the elevator and from the end of it the present shuttle conveyor will take coal for delivery to the new bin. In this way a 70-foot

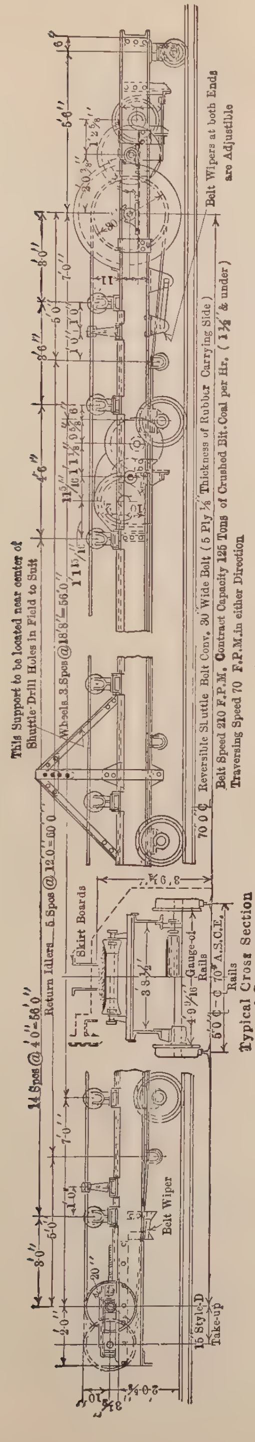


FIG. 170.—Side View and Cross-section of Shuttle Conveyor Shown in Fig. 169.

shuttle conveyor will serve 208 feet of bin. Fig. 170 shows the side-view and cross-section of the conveyor and its frame.

When the full length of a shuttle conveyor is always over a bin the spill of fine material from the return belt drops into the bin, but when a part of the conveyor extends beyond the bin, spill of fine material onto the floor below may be objectionable. In such a case a belt cleaner or brush should be used at one or both ends of the conveyor. The belt shown in Fig. 170 was cleaned by a wiper at each end, a piece of rubber belting backed up by a steel strip and set diagonally under the return run, just touching the conveyor belt. It was adjustable for wear and was simpler than a revolving brush.

CHAPTER IX

PROTECTING AND CLEANING THE BELT

Material Adhering to the Belt.—When coal, coke, clay and ores are handled particles often cling to the belt after passing the discharge point and then fall off on the return run on meeting the idler pulleys. If the return idlers are close to a floor or located near the framing of a bent, as in Fig. 171, dirt piles up so as to prevent the pulley from turning; the belt may then wear a hole in the rim of the pulley and may be cut and perhaps ruined. This is more likely to happen if the conveyor is enclosed in a box-like housing, or if the return idlers are below a foot-walk, hard to get at, and hence the grease cups are not filled and screwed down regularly. It often happens,

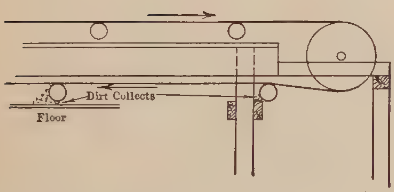


Fig. 171.—Spill of Dirt on Return Run.

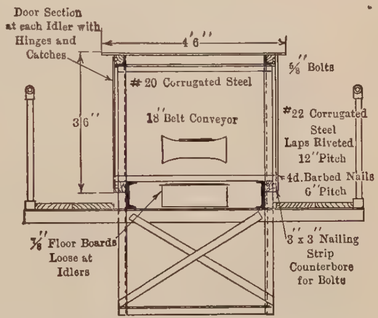


Fig. 172.—Steel Bridge and Enclosure for Belt Conveyor. Housing Open on the Bottom.

also, that the return run is hidden from view by the protective deck. In Fig. 172 the return run is hidden by the deck, but the gallery is open under the return belt to let dirt fall away. If the floor cannot be omitted, there should be plenty of space below the return belt so that the spill can fall clear of the idlers and be easily seen and removed.

For the same reason, it is well to set return idlers in relation to the framing of bents and the travel of belt so that the dirt falls clear of the bent, as if the belt shown in Fig. 183 ran in the opposite direction.

Cleaning Devices.—In many cases the dribble of material along the return run of a belt conveyor is not serious, but in other cases it is so objectionable that a cleaning device must be used near the head pulley to remove clinging particles from the belt. It is also necessary to clean the belt when snub pulleys or bend pulleys or tandem-drive pulleys make contact

with the dirty side of the belt on the return run; it not only prevents particles from being forced into the belt by the pulleys, but it also prevents material from accumulating on the rims of the pulleys and forming crusts there which cause the belt to run crooked, or perhaps injure it.

Stationary brushes are not a success for cleaning belts; they fill up with dirt and fine stuff and soon become useless. Air blasts have been used, but they require air under pressure, and the cost of operation is great. Strips of belting set diagonally against the under side of the return belt have been satisfactory on some belts handling coal.

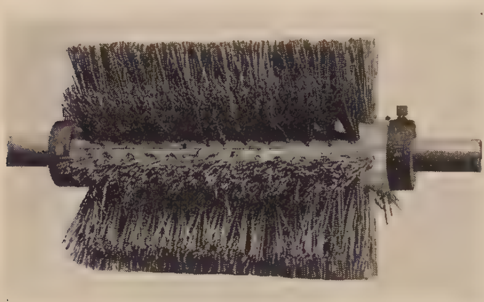


FIG. 173.—Revolving Brush for Cleaning the Belt.

Revolving Brushes.—A revolving brush (Fig. 173) consists of bunches of rattan or fiber splints glued into holes drilled in a wooden cylinder. Fig. 174 shows one with its drive; it must be set so that when the loaded conveyor is started at slow speed, the material falling vertically from the head pulley will not hit the brush. A chute to collect the scatter and spill back of the brush is not always necessary and should be avoided if possible,

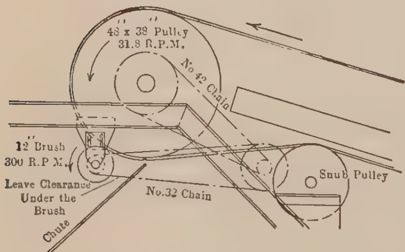


FIG. 174.—Head of Belt Conveyor and Drive for Cleaning Brush.

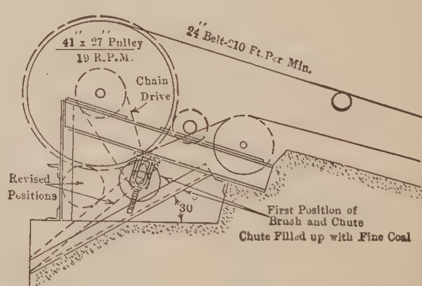


FIG. 175.—Right and Wrong Positions of Cleaning Brush and Drip Chute.

but when it is used, the angle must be steep enough to let damp or sluggish material flow readily and there should be room enough between brush and chute to avoid clogging. Fig. 175 shows a 24-inch belt discharging crushed coal to a grab-bucket pit. In its first position the chute clogged; when the brush was moved forward and the chute was made steeper it worked well.

Speed of Brushes.—To be effective, a brush must work *against* the travel of the belt and at a speed sufficient to throw the fine stuff out of the bristles and keep the brush clean. Some brushes which do not work well have the bristles or bunches of splints set too close together or they are

run at speeds that are too low. The speeds in feet per minute at the tips of the bristles should be, for brushes 8 to 12 inches in diameter:

Dry materials	800 to 1000 feet per minute.
Damp materials	1000 to 1200 feet per minute.
Wet and sticky materials	1200 to 1500 feet per minute.

The brush should be mounted so that it can be adjusted toward the belt to compensate for wear on the tips of the bristles and in such a way that the drive to the brush is not affected (see Fig. 174). A revolving brush does proper work only when it is kept in proper adjustment toward the belt. On that account and because at the high speed necessary brushes are often short-lived, some users of belt conveyors do not employ them, but let the dirt fall away from the belt and then clean it up regularly.

Other Belt Cleaners.—Since it is not easy to repair a fiber brush with the materials and labor ordinarily available a number of substitutes have been used. Ridgway, in 1912, patented a belt beater which consists of a shaft carrying several pivoted arms on which are loosely mounted pipes extending across the belt. Under the action of centrifugal force, the arms assume a radial position and the pipes strike a glancing blow against the belt. It was designed to work under a free stretch of belt, not against a belt in contact with a pulley. It is not in use now.

Fig. 176 shows a belt flapper which has some resemblance to Ridgway's device. It is a wood cylinder to which four or five strips of old belt are screwed. It is cheap, easily made and readily repaired; it is to some extent self-adjusting and will clear itself of fine stuff at speeds less than those given for bristle brushes. The outer edges of the strips are sometimes weighted by steel flats. The Winters patent of 1920 covers a belt cleaner with rubber strips similar to Fig. 176 but mounted beneath the return belt on a frame with screw adjustment horizontally and vertically. The vertical adjustment compensates for wear on the edges of the strips and the other permits the drive from the conveyor head shaft to be kept at the proper length center to center.

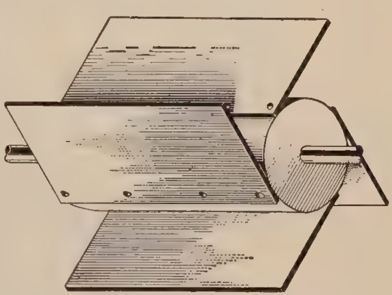


FIG. 176.—Rotary Flapper to Clean a Belt.

Other belt cleaners have been used experimentally. One is a repair for an ordinary brush; when the rattan bristles wore out, they were replaced by strips of old belt bent in U form and screwed to the wooden cylinder in a helix (spiral) of rapid pitch so that two or three of the edges of the strips were always in contact with the belt. Another scheme consists of disks of old belting 12 inches in diameter strung on a shaft at a slight angle and nailed to oblique sections of a wood cylinder which support the disks

and act as spacers. This is said to work by frictional contact with the belt and to outlast three ordinary rattan brushes, but like many other "kinks" it is apt to work better for the inventor than for anyone else.

The Carr patent of 1920 covers a method to remove, from the return run of a belt conveyor, wet, sticky material like cement mortar. The return belt is kept tight and by means of a revolving cam-shaped roller pressing down on it, the belt is caused to whip or snap violently and shake off the adhering particles.

Sprays of water have been used with success to clean belts handling wet concrete.

The idea of cleaning a belt by leading the return run across a wide nozzle which forms the inlet of a vacuum cleaner is disclosed in the Bemis patent of 1918. It is designed to remove dust and dirt from the canvas belts used for carrying packages in stores.

Tripper Brushes.—Trippers that handle sharp, wet material like sand or crushed ores should have a brush just ahead of the lower pulley to prevent

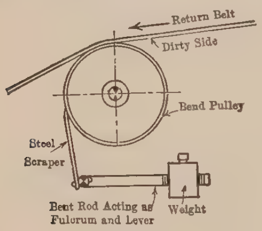


Fig. 177.—Pulley Rims Kept Clean by Use of a Scraper.

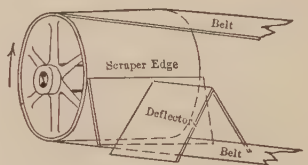


Fig. 178.—Scraper for Pulley Rim Combined with a Deflection.

sharp particles from being forced into the belt or its cover on the reverse bend.

Cleaning Pulley Rims.—It is sometimes necessary to use steel scrapers on the rims of snub pulleys, tripper pulleys and deflector pulleys in order to prevent the accumulation of material which might hurt the surface of the belt or cause it to run out of line. Fig. 177 shows a scraper fitted to a weighted lever and Fig. 178 illustrates a deflector to shed the scrapings clear of a lower run of belt.

Protective Deck.—In order to prevent scattered material from dropping onto the return belt it is advisable in most cases to cover the space between the conveyor stringers by a floor or deck of plank or light sheet steel. Fig. 172 shows such a deck in cross-section (see also Fig. 163). It lessens the risk that lumps of material or sticks or tools or similar things falling on the return belt may be carried between the foot pulley and the belt and perhaps tear it or punch a hole in it. On the other hand, it affords a lodging place for dirt, and on inclined conveyors, lumps falling on the deck may lodge against the idler pulleys in such a way as to form a very effective brake to prevent rotation. In Fig. 163 a guard has been placed across the deck to prevent spill from jamming against the hump pulley.

The deck may be omitted from conveyors that carry only fine material and from inclined conveyors where there is a tandem drive or a bend pulley near the lower end of the return run. At such places the lumps can be deflected or thrown off the belt without doing any damage.

When belt conveyors are mounted on elevated frames or bridges, or enclosed in housings, the deck is sometimes objectionable because it covers up the lower run out of sight and hinders or prevents access to the bearings of return idlers. In such cases, if the conveyor belt is plenty wide enough and not likely to spill over the sides, the deck may be omitted and a wiper or plow used over the return belt near the foot to push off any stray piece before it comes to the pulley. Doing this is, of course, choosing between two evils; the right way is to use a deck and place the conveyor stringers so that the return belt is visible for inspection (see Fig. 182).

In some cases where belts are loaded at a number of places along their length as on coke wharves at by-product plants, a deck does not afford complete protection against lumps getting on the return belt and is a place where coke may lodge and do damage. At one plant the return belt was run over two tripper pulleys near the foot wheel. This put two extra bends in the belt but it prevented lumps of coke from being jammed between the belt and the foot pulley, and it gathered the spill at one place convenient for removal (Fig. 179). There was no deck over the lower run in this case.

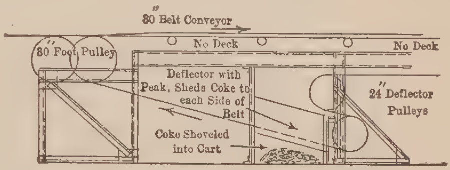


FIG. 179.—Tripper Device to Remove Lumps from Return Run of Belt. (R. H. Beaumont Co.)

Belt Conveyor Enclosures.—When a belt conveyor runs outdoors throughout all its length, or from one building to another, it may be necessary to cover it for several reasons: (1) To avoid exposure to sunlight, rain or snow; sunlight is injurious to rubber (see p. 44); the material carried on the belt may be damaged if wet, or the belt may carry water into a building or into a bin and create a nuisance. (2) To avoid exposure to wind; a strong wind may blow material off the belt or may lift the belt off the idlers or cause it to run crooked.

The simplest enclosure is merely a roof over the conveyor, with the sides left open. This construction on an open wood deck-bridge is similar to that shown in Fig. 180. The roof is made of corrugated steel sheets screwed on, and the dimension A is made about 18 inches so that when the sides are left open a man can reach in under the roof to the grease cups of the troughing idlers on both sides of the conveyor. The return idler pulleys in this design run loose on the shaft and are lubricated from one large cup on the footwalk side. When sides are added, the sheathing on the footwalk side is held on by bolts with thumb-nuts; that on the far side is nailed on. Over each troughing idler there is a removable door in the roof sheathing about 24 inches wide, to give access to the grease cups of the troughing

idlers and dimension A is made about 12 inches to avoid too long a reach.

Fig. 172 shows a steel deck-bridge with two footwalks that give access to bearings on both sides through hinged doors in the corrugated sheathing opposite each idler. Bearings for return idlers are reached by removing loose boards from the decking above the lower run, but in this design, as in the one shown in Fig. 180, the return belt and its idlers are practically concealed from view. This is always an objection; if an idler pulley sticks and refuses to turn, it may damage the belt if it is not detected promptly.

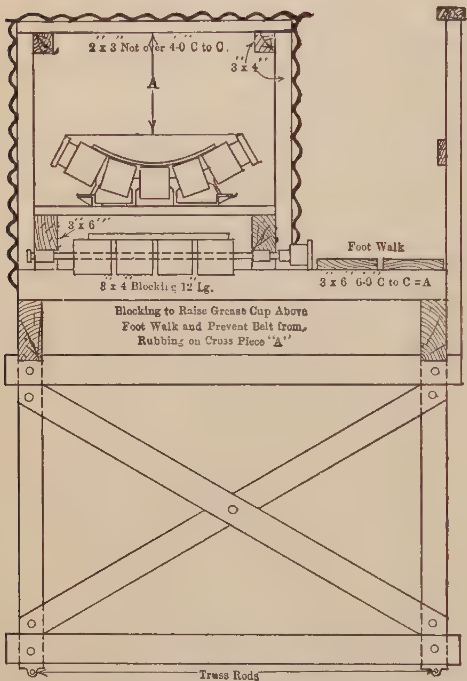


FIG. 180.—Enclosure for Belt Conveyor, Corrugated Steel on Wood Frame. (R. H. Beaumont Co.)

Fig. 181 shows an 18-inch belt conveyor enclosed in a through-bridge. The structure was made wide enough for a tripper, but there was a footwalk on one side only and it was not easy to get at the far side of the tripper for lubrication and the necessary inspection and attention. Another defect in the design is that the return run is completely covered. It is hard to get at the bearings and the belt can be seen only by unbolting and removing sections of the steel deck.

A design better in both respects is illustrated in Fig. 182. Both sides of the conveyor are accessible for inspection and lubrication, and the return belt can be seen on both top and bottom for its full length. The bridge is open below the return run so that dirt falls away clear of the conveyor, and if any does lodge on the lower bracing of the bridge, it cannot foul the return idlers and prevent them from turning (see Fig. 171).

A belt-conveyor housing used by the Stephens-Adamson Manufacturing Co. on open deck bridges is in section an inverted U, open at the bottom and wide enough to cover the belt, the idlers and the stringers on which they are mounted. One panel-length of housing covers two troughing idlers; transverse slots in the rounded top provided with doors give access to the idlers for examination and lubrication.

It is worthwhile to emphasize the statement that when belts handle coal, coke and other substances which are regularly or at times moist, some

of the fine particles cling to the belt and are dropped off at each idler on the return run, most of them near the discharge end, but often at each idler back to the loading point. This spill is often a serious matter, and if it cannot be allowed to fall freely away at each idler, the supporting structure

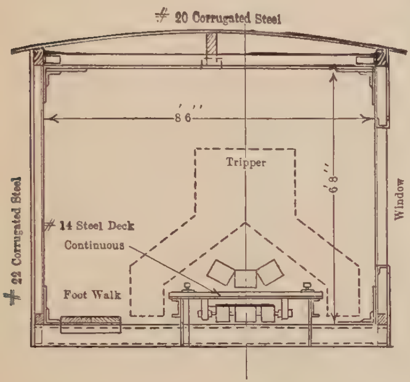


FIG. 181.—Defective Design of Steel Bridge and Belt Supports.

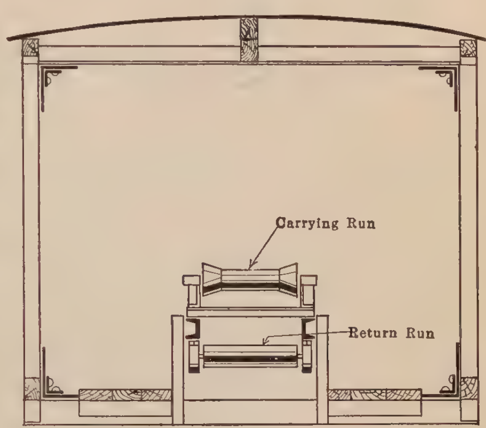


FIG. 182.—Bridge and Supports for Belt Conveyor. Both Runs Visible and Accessible.

or enclosure should be designed so that the spill can be removed regularly and easily.

Enclosures which are designed chiefly for architectural effect and neat and trim appearance often cause trouble and expense, because they confine the spill and hinder inspection and cleaning.

CHAPTER X

PACKAGE CONVEYORS

Package Conveying.—The belt is generally the cheapest and best conveyor for packages of light weight, papers, books, wrapped goods, sacks, bags, and for boxes that are not too heavy. The belts are always run flat.

Belts for Package Conveyors.—Since the work is usually dry and indoors, and the material not harmful to the surface of the belt, it is not necessary to use a belt with a high resistance to abrasion and to the action of water. Many package conveyors that carry light goods use solid-woven belts (see p. 50) of a thickness corresponding to 4-ply. If the length is short, the belts may be used just as they are woven, without waterproofing to resist atmospheric moisture; they are cheap, clean, and contain nothing to mark or stain the goods carried. Such belts are very flexible and will bend readily over the 8-inch end pulleys and the $2\frac{1}{2}$-inch idlers generally used in such conveyors. In longer conveyors where the stretch of an untreated belt is objectionable, fabric belts (solid woven or built-up) may be treated with a colorless Class 3 compound (see p. 48). This is clean, and leaves the belt quite flexible. Belts for handling baskets, boxes, heavy parcels, mail bags, express matter, etc., need a density of body and a surface toughness to resist the bumps, blows and scratches caused by sharp corners, nails and metal fastenings. They are frequently stitched canvas belts with Class 1 impregnation (see p. 48), not so flexible as those treated with Class 3 compounds, but more resistant to wear. Fig. 183 shows one of many such belts used in a Chicago mail-order house. Table 9, page 53, gives a comparison between two specimens of solid-woven belt and one stitched canvas belt as to stretch and ultimate strength.

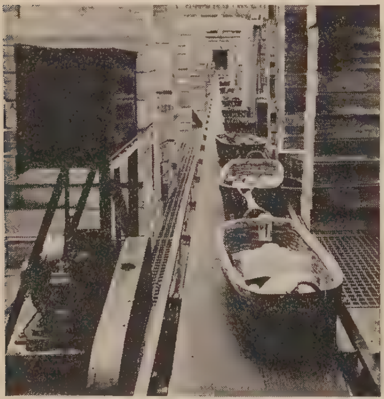


FIG. 183.—Stitched Canvas Belt for Assembling Goods on Mail-orders. (Imperial Belting Co.)

Package conveying does not often require a belt with a minimum amount of stretch or a maximum resistance to moisture; hence rubber belts and balata belts are not often used for this work unless their prices are

low enough to compete with solid-woven or stitched belts. When rubber belts are used a cover is seldom needed, experience showing that friction surface belts resist the wear sufficiently well and that cheaper belts are more economical in the long run.

Supporting Rolls for Package Conveyors.—Wood rolls are apt to warp, crack and get out of balance; they are practically obsolete. Steel rolls in sizes 4 inches and larger are generally made of steel pipe or tubing shrunk onto cast-iron heads in which are inserted shafts, either as short stub ends

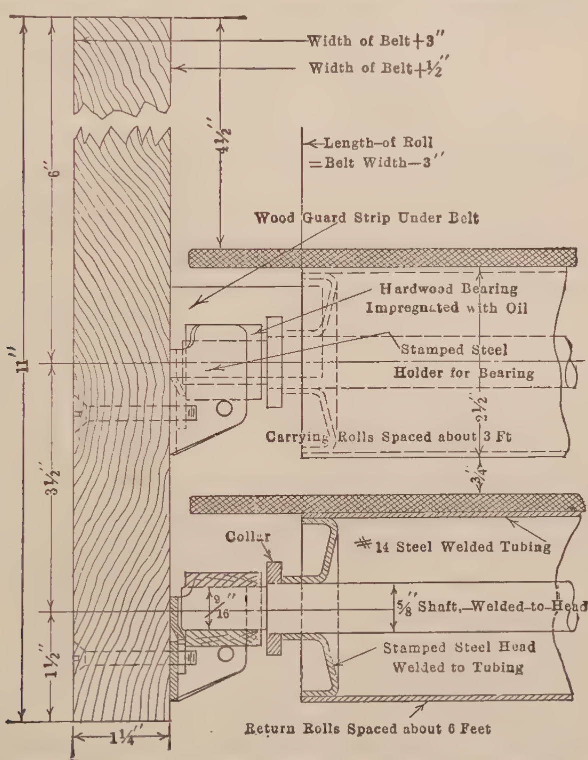


FIG. 184.—Package Conveyor, Wood Frame and Oilless Bearings. (Lamson Co.)

or "gudgeons" cast with the heads or set screwed in place, or as shafts extending through both heads with projection enough for a bearing at each end. For light belts, it is important to have a roll that is light, lively, and accurate in balance; most of them for store service are made of 2½-inch diameter steel tubing. If stub-end shafts are used, the cast-iron heads must be of sufficient length to maintain the tube and the two end shafts in permanent alignment (see Fig. 186). If a through shaft is used, the heads may be much lighter, but the shaft must be perfectly straight (see Fig. 184).

Most package conveyors are used indoors and in places where neatness

and compactness are of some importance. Fig. 184 shows a cross-section of both runs of belt conveyors from 6 to 48 inches in width, all contained within the depth of a 11-inch plank side. The edges of the belt overhang the ends of the rolls by $1\frac{1}{2}$ inches, but are prevented from sagging too far and dropping goods by a wood guard strip which is continuous for the length of the conveyor except for clearance at the bearings. Fig. 185 shows a similar construction used for packages in stores where the conveyor is hung from the ceiling by rods, and where for the sake of appearance it is enclosed on sides, bottoms and ends with wood construction and wall-board panels. The bearings shown in Fig. 184 are hard wood blocks impregnated with a lubricant and held in a yoke which allows them some freedom of movement. Such bearings require no oil; they are sufficient for moderate loads. The bearings shown in Fig. 185 are bronze, mounted in cast-iron holders in such a way as to maintain alignment with the roll shafts.

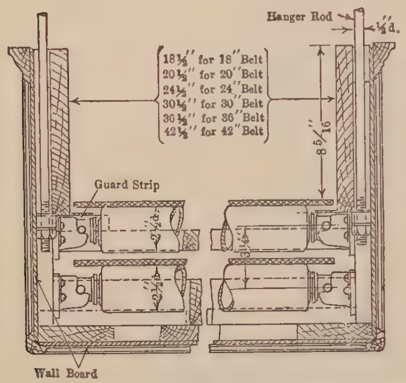


Fig. 185.—Package Conveyor, Panel Board Enclosure. Bronze Roll Bearings. (Lamson Co.)

Fig. 186 illustrates a 42-inch belt conveyor with all steel construction;

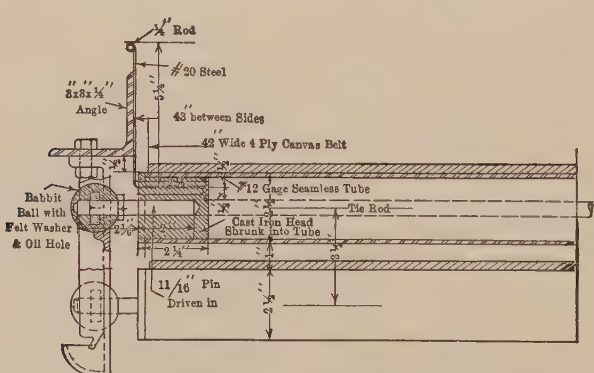


Fig. 186.—42-inch Belt Conveyor for Department Store Service. (Link-Belt Company.)

the shaft bearings are spherical balls of Babbitt metal cast with a central chamber in which is fitted a felt washer to retain oil for lubrication.

Other Forms of Package Conveyors.—Bulletins of manufacturers show package conveyors with the belt supported on one or both runs by oiled wood strips instead of rolls; conveyors with upper run and lower run spaced apart so that they can be loaded with goods on both runs for move-

ment in either direction; conveyors for trays where the belts are carried on rolls set close together.

Capacities of Belts in package conveyors are much less, measured in pounds, than the corresponding capacities in handling bulk materials. The service is generally intermittent, the goods are usually placed on by hand, the speed is low—75 to 150 feet per minute, and the width is often determined by the dimensions of the packages rather than by the quantity to be handled. The rate at which the packages can be taken away at the discharge point will sometimes determine the carrying capacity of a belt.

Discharge is generally over the end pulley or by means of scrapers at

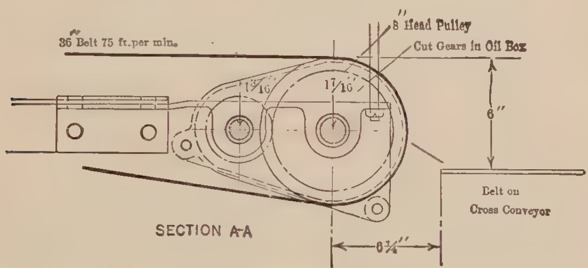


FIG. 187.—Transfer Between Two Package Conveyors. (Link-Belt Company.)

intermediate points. A typical transfer between two package conveyors in a department store is shown in Fig. 187. The head pulley is 8 inches in diameter; it is placed quite close to the receiving belt with merely a guard to bridge the few inches of gap. This short connection saves height, lessens the chance of breakage of goods and prevents an accumulation at the transfer point. In transferring sacks and bulky packages, it is better to place the receiving belt lower, so that the package has more drop and will be taken away promptly; otherwise it may hang between the two conveyors and be rubbed and perhaps torn by the moving belts.

CHAPTER XI

SPECIAL USES OF BELT CONVEYORS

Special Machines Using Belt Conveyors.—Within the past ten years a number of portable belt-conveyor machines have come into use for loading and unloading cars, piling bulk material into storage or reclaiming from storage and doing similar work formerly done by human muscle with shovel and wheelbarrow. In order that these machines can be easily handled and moved from place to place the frames are made as light as possible and the conveying apparatus and the driving machinery have been designed especially to be compact and without superfluous weight.

The portable conveyor made in several styles by the Barber-Greene Co. of Aurora, Ill., consists of a standard discharge end, a standard drive end with a loading hopper and an electric motor and standard intermediate

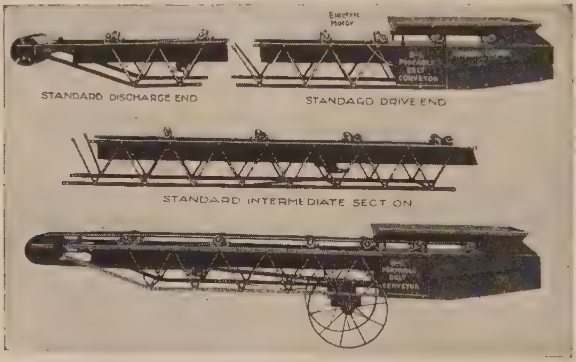


FIG. 188.—Portable Steel-frame Belt Conveyor. (Barber-Greene Co.)

sections (Fig. 188). When the two end sections are coupled together and mounted on a pair of wheels they make a portable machine 15 feet long. The intermediate sections are made in lengths of 3, 6, 9, 12 or 15 feet; where these are inserted between the standard end sections they make portable machines of various lengths up to 60 feet. The belts are 12, 18, 24 or 30 inches wide, and run over three-pulley idlers set at 30° troughing. The pulleys are made of steel tubing.

The Scoop Loader (Wentz reissue patent of 1920), made by the Portable Machinery Co. of Passaic, N. J., is especially light and portable. The belt is 12 or 16 inches wide and has cleats or flights to prevent material from

slipping back; it runs flat between continuous skirt-boards with most of the weight carried on small rollers, but with its edges supported on wood stringers (Fig. 189). The lower end of the frame is brought to a sharp point and has a small belt pulley set very low so that the foot of the machine

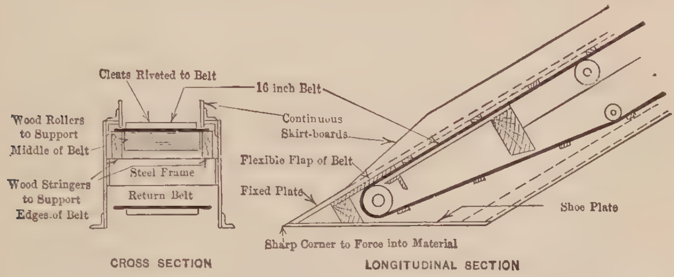


Fig. 189.—"Scoop" Loader, Bottom end and Cross-section. (Portable Machinery Co.)

can be forced into a pile of material and pick up its load without much shoveling. Judged by usual belt-conveyor standards, the pick-up of material is bad and the support of the belt inferior; but in the service for which the machine is generally sold, the work is intermittent or occasional,

Fig. 190.—Pratt Loader Piling Sand in Box-car. (Link-Belt Co.)

and the belt lasts long enough to show a low cost per ton of material handled. The wear on the belt is often a minor consideration in machines of this general class, they enable one man to do the work which formerly took two or three; they do the work more quickly, reduce charges for car demurrage and often increase the available storage capacity of a yard or a shed. To

get these advantages with simplicity of construction and ease of handling the machines it may be true economy to let belts wear out rapidly.

The Pratt Loader (Fig. 190) (Link-Belt Company, Philadelphia) is a short belt conveyor mounted on a light wheeled frame and run so fast that it throws the material far beyond the end of the machine. A machine standing in the doorway of a box car can pile material high at the ends of the car. The belt is fitted with cleats and slides at 500 feet or 1000 feet per minute over a steel bottom plate between continuous skirt-boards on the upper run. Naturally the belt does not last so long as it would in an ordinary belt conveyor, but as has been said above, the durability of the belt is subordinated to other considerations in apparatus of this kind.

The Manierre Loader (Manierre Engineering and Machinery Co., Milwaukee) is a belt conveyor mounted on a steel frame hinged at several points so that it will swing into a box car through the doorway. The conveyor does not run fast; it is adjustable for angle and will pile material into the ends of the car without breakage, starting with the delivery end close to

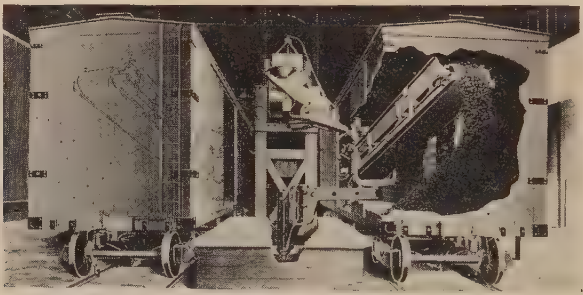


FIG. 191.—Manierre Loader Receiving from Belt Conveyor and Delivering to Box-car.

the floor and raising as the pile forms. It is used chiefly for material like coal and coke which must be handled without unnecessary breakage (Fig. 191).

Distributing Coal in Holds of Ships.—At shipping piers at Greenwich Point, Philadelphia, Curtis Bay and Port Covington, Baltimore and elsewhere, coal is distributed in the holds of ships away from hatches and close up under decks by means of high-speed belts attached to the lower end of telescopic chutes (see Fig. 167). These belts are 36 or 48 inches wide, run at 2500 or 2700 feet per minute over pulleys 12 or 18 inches in diameter set about 4 feet apart and threw the coal 30 or 40 feet beyond the machine at the rate of a carload (50 tons) in two or three minutes. Fig. 192 shows the assembled lower end of the telescopic chute with men putting on a new belt. The belts are usually 8-ply rubber or balata belts with strips of the same material about $1\frac{1}{2}$ inches wide riveted across the carrying surface every 6 inches and with similar pieces riveted along the margins to take the edge wear. At the very high speed, the abrasion of the carrying surface is severe and rapid, rubber covers do not last long and the life of the belt

is determined by the rate at which it is weakened by the destruction of its plies of fabric. A canvas or balata belt with a heavy close-woven duck does not give quite the service of a high-grade rubber belt with a thick cover, but it is sold at a lower price and in some instances carries enough coal to make it more economical than the rubber belt. Good balata belts carry about 30,000 tons on one of these distributors; expensive rubber belts do not average 40,000 tons. This represents only a few days or weeks of life, nevertheless it pays to wear out belts at $100 or $200 apiece rather than employ gangs of shovelers to trim the coal in the cargo space of the ships. The loading is done better and in much less time; and the ship is not held so long at the pier and out of carrying service.

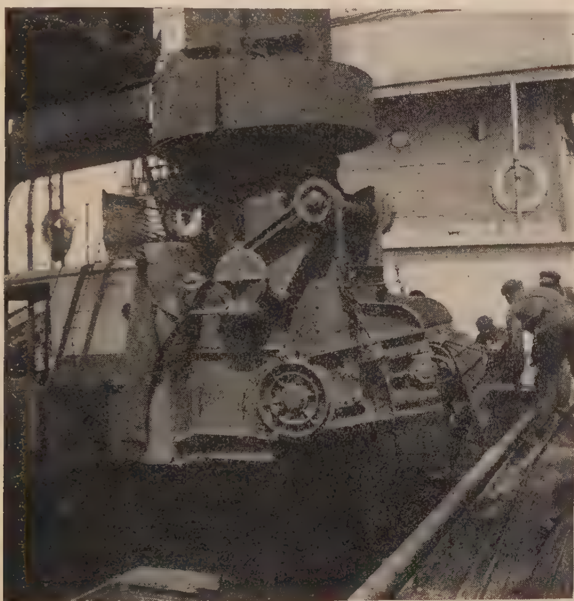


Fig. 192.—Lower End of Ship-loading Chute Fitted with High-speed Belt for Trimming Coal Under Decks.

Conveying between Two Belts.—The angle at which a belt will carry material up an incline is limited by the tendency of the material to roll back on itself or slip on the belt, but if the material can be prevented from doing that, it can be carried at a steeper angle. It can be done if a second belt travels along with the conveyor belt with slack enough to lie over and confine the material.

This principle has been used in agricultural implements to elevate straw, etc., and even to carry coal in inclined belts forming part of loading machines, but as applied to bulk materials which consist of a mixture of lumps and fines, the hold of the upper belt is uncertain and some of the material

may slip back or fall out. This drawback does not exist to the same degree with packaged goods, and when the pieces carried are uniform in size and not too thick, like newspapers, there is no difficulty in conveying them at angles up to the vertical.

Newspaper Conveyors and Elevators.—The original newspaper conveyor of this type (Perkins patent, 1880) is shown in Fig. 193. By confining the vertical runs of the two belts A and B between pulleys set opposite or staggered, the papers are prevented from slipping by the pressure of the belts toward each other. There is, however, one objection to the use of belts for this work, when the machines take papers directly from the press, as is usually the case. The ink is then not quite dry and will be smeared or smudged if the belts absorb the ink or if in passing over the pulleys there is movement between the belts and the papers. In the most recent machines of this type, built by the Lamson Company, the belts have been replaced by cotton cords wound round with a covering of soft iron wire (Cowley patent, 1917). The wire touches the papers in small spots only and is not likely to smear the ink.

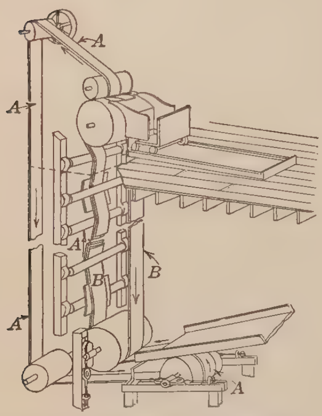


Fig. 193.—Elevating Newspapers Between Two Belts.

The same principle has been used (Lamson Co.) in a machine to handle tin crowns or bottle caps. The caps are carried between two belts which are grooved (Fig. 194) to such a depth that when the belts are together,

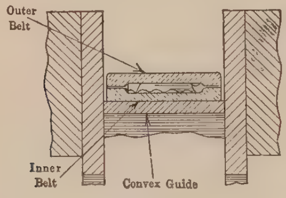


Fig. 194.—Grooved Belts for Elevating Small Articles. (Cross-section of Vertical Run.) (Lamson Co.)

Fig. 195.—Conveying Material between Two-flanged Belts.

the caps will be squeezed between them. On the vertical run, the belts are guided over a surface slightly convex so as to keep them together and prevent the caps from slipping.

German patent 52697 to Luther illustrates the same idea applied to bulk materials (Fig. 195). It is not in practical use.

The Anderson patent of 1920 proposes to use two short vertical belts placed under the hopper of a molding machine as a means to draw sand from the hopper and throw it forcibly into a flask or sand-mold, so as to pack the sand and lessen the labor of ramming.

Belt Conveying by Rolling Contact.—The Hopkins and Fellows device (patented 1904) used in can factories and canneries to carry and elevate round tin cans is shown in Fig. 196. Cans rolled down the guide at the foot of the machine rest on a yielding section of track and come in contact with the elevating belt. This is kept under tension by a weighted pulley and by being deflected from a straight line by the convex curvature of the track on which the cans roll. The motion of the belt rolls the cans up the track until the track ends as shown in the figure, or the track may be curved with a flexible section around the upper pulley to return the cans to the side from which they came.

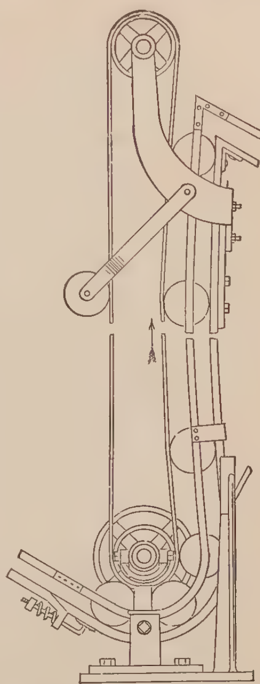


FIG. 196.—Conveying and Elevating Tin Cans by Rolling Contact.

Picking or Sorting Belts.—When belts are used to expose ores or minerals to the inspection of men or boys who remove waste material from them the travel must be slow, generally not over 50 feet per minute and even less if the material is lumpy. The bed of material should be thin and wide so that all the pieces can be seen and can be turned over without too much exertion. A belt wider than 48 inches makes the pickers (boys especially) reach too far; they cannot work so well, and the sorting is not so efficient as when the belt is 36 inches wide.

Capacity.—The capacity of a picking belt is an indefinite quantity, but it is always less than the capacity of the belt calculated as a conveyor for its particular speed. The following may be used as representing the working capacities of picking belts on metalliferous ores.

Capacity of 30-inch belt on three-pulley idler with broad center pulley in cubic feet per hour $=1.3W^2$ at 50 feet per minute.

Capacity of 36-inch belt on three-pulley idler with broad center pulley in cubic feet per hour $=1.1W^2$ at 50 feet per minute.

Capacity of 42-inch belt on three-pulley idler with broad center pulley in cubic feet per hour $=1W^2$ at 50 feet per minute.

Capacity of 48-inch belt on three-pulley idler with broad center pulley in cubic feet per hour $=0.9W^2$ at 50 feet per minute.

Idlers for Picking Belts.—Standard three-pulley or five-pulley idlers are not generally used for this work; they crowd the material toward the center of the belt. It is better that the belt should run nearly flat with the material spread out to near the edges; at the slow speed there is not much tendency for the pieces to roll off. Three-pulley idlers (see Fig. 70) with a broad-face center pulley are listed for this work by several manufacturers; they spread the material out better than standard idlers. Spool idlers

(see Fig. 84) or flat rolls with occasional concentrators (see Fig. 28) can also be used.

The number of men required to sort or pick depends upon the size and weight of the pieces and the amount of material to be removed from the belt; it can be determined only by a test under operating conditions. Peele's

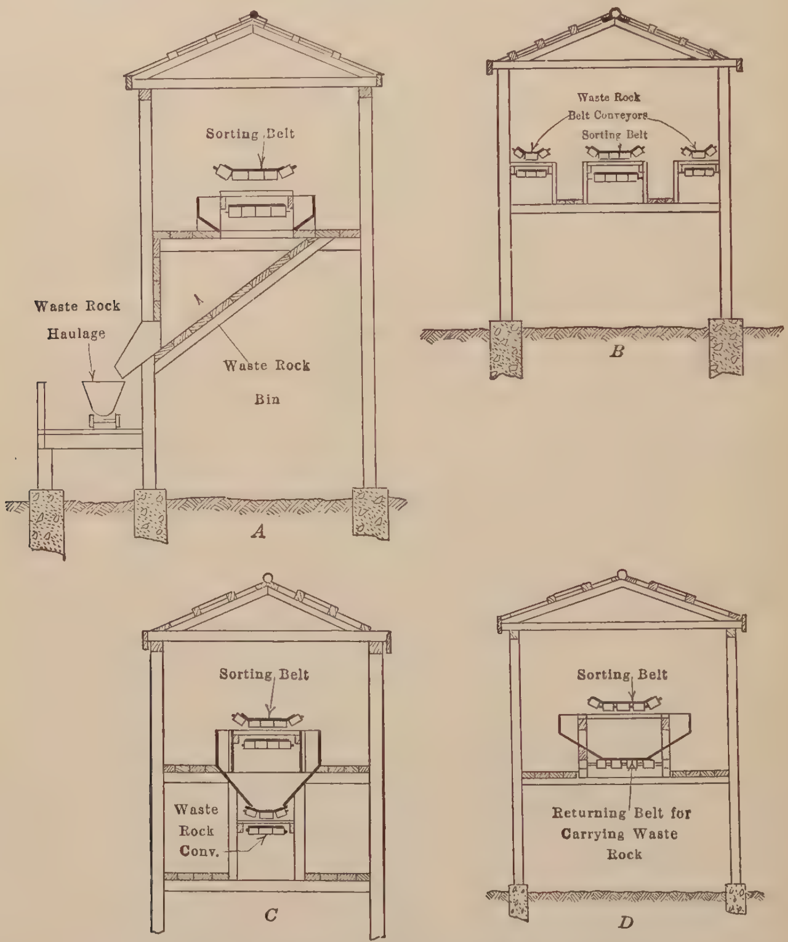


FIG. 197.—Various Arrangements of Picking Bands for Disposal of the Waste.

Mining Engineers' Handbook, 1st edition, page 1651, says that the weight per hour picked off a belt in 1-inch pieces weighing $\frac{1}{4}$ pound is about one-fifth the weight of the material picked off in 6-inch lumps weighing 58 pounds and that the maximum is attained if the pieces are about 3 to 7 pounds weight.

The length of a picking belt is determined by the number of pickers

and by the space alongside allowed for each one. Thirty to 40 inches is usual per man on each side of the belt.

The incline should not exceed 10° or 12° to prevent lumps from rolling on the belt and hurting men's hands. Moreover, men cannot stand comfortably on an incline; if the belt is on a slope it may be better to build the working platform as a series of steps; for a 10° slope, a 36-inch tread has a 6½-inch rise.

Disposal of the Waste.—Fig. 197 (Schmitt's Text Book of Rand Metallurgic Practice) shows four methods. The scheme of Fig. *B* allows two kinds of separation to be made, the pickers on each side of the belt picking off one grade of material as waste or for retreatment. It requires the pickers to turn around, and in that way they lose time. Fig. *C* is a common arrangement; it is easy to drop the pickings into the chute boxes, one at each man's place, and the main belt when worn can be cut down or pieced to make the lower belt: The disadvantage of Fig. *D* is that the discharge from the lower run requires two pulleys to be arranged as a tripper near the foot similar to Fig. 179. The pulley side of the belt, not generally protected by a cover, is apt to be injured by particles of rock adhering to it and forced into the fabric by contact with the pulleys.

In South African practice the work on the picking belts is very severe, and in spite of their slow speed the belts do not have a long life. The quartz ore handled there breaks with very sharp fractures and is apt to cut the belt by impact at the loading point, and when the pickers drag the heavy pieces over the belt and over its edges. In some plants a part of the conveyor near the foot is set on an incline and the ore is washed there by water sprays. The combination of hard, sharp ore and water causes wear on both sides of the belt and on pulleys and idlers as well.

CHAPTER XII

LIFE OF BELTS

Life of Belts.—The life of a belt depends upon many factors. A full discussion of them would be a review of what has already been said about belts and their accessory parts. We may put down in a list a number of things which tend to shorten the life of a belt, see page 197.

The belt salesman is often called upon to answer the question, "How long should this belt last?" With some knowledge of the items in Table 32 a prudent man will say that he cannot tell, that he is in the position of an actuary of a life-insurance company when asked how long a certain policy-holder will live. It is possible to quote tables of average expectancy of human life; it is also possible to say, from a knowledge of some hundreds of belt conveyors, that in handling material like coal, not too heavy and not too abrasive, a rubber belt W inches wide with $\frac{1}{8}$-inch cover on a conveyor 100-foot centers should not wear out before it has carried $500W^2$ tons, and on conveyor 200-foot centers, twice as much, etc.

This rule, if it may be called a rule, is based upon considerable knowledge and experience in making and selling rubber belts, but it is not based upon a sufficient number of cases nor has it been under consideration long enough to be called authoritative. It is never offered as a guarantee by the concerns that quote it, nor is it ever mentioned in considering belts for coke, ores and minerals. Its application to any specific case, even for coal conveying, is so conditioned and so hedged about with contingencies that it is really not safe to offer it as an answer to the question cited above or bring it into the discussion of a particular conveyor.

The following table (Table 33) gives cases taken from actual practice and compares the actual tonnage handled with the tonnage given by the statement, Tons $= 500W^2$ per 100-foot centers.

Typical Injuries to Belts.—Companies in the business of making belts or selling belt conveyors are accustomed to receive complaints about the quality of belts somewhat in this style: "The belt we bought from you on January 1st has lasted only six months and is going to pieces. Our previous belts all lasted two years; we think you should furnish a new belt or make some allowance on the price of the one we bought, etc., etc." Where it is possible to investigate such complaints, it will generally be found that the cause is one of the twenty-eight items given in Table 32. For some of these, the seller of the belt may be responsible, but more often he is not.

TABLE 32

1. Buying a cheap belt—poor grade of rubber or light-weight fabric.
2. Buying an old belt—rubber dried out, no elasticity.
3. Injury to belt by carelessness in getting it into place on the conveyor —cover or edges torn.
4. Splice not square with the belt—belt runs crooked, wears edges.
5. Material too hot—burns holes in the belt, chars the cotton.
6. Material corrodes cotton or softens rubber—plies come apart.
7. Loading chute of poor design—material wears the belt surface.
8. Skirt-boards too long or badly set—belt cut.
9. Loading belt directly over idler—belt cut.
10. Speed too fast for proper pick-up of lumps—belt cut or abraded.
11. Speed so fast that belt carries only thin load—belt wears out in center.
12. Cover too light for sharp lump material—belt cut, cover worn.
13. Handling wet, sharp material through a tripper—cover cut and worn.
14. Pulleys too small at drive, take-up, bend, snub or tripper—plies come apart.
15. Wide gaps between idler pulleys—belt cracks longitudinally.
16. Troughing too steep for thickness of belt—belt cracks, also runs crooked.
17. Bad alignment of conveyor—edges rubbed off, or worn against side-guide idlers.
18. Too many side-guide idlers—wear off edges of belt.
19. Too many fixed trippers—belt injured by reverse bending and by putting material back on belt.
20. Excessive take-up tension—belt stretches and splices pull out.
21. Reverse bends in belt—plies come apart, wear on pulley rims and belt surface.
22. Exposure to snow, rain, sunlight—rubber decays.
23. Lubrication neglected—belts cut and torn by worn idlers.
24. Frequent starting under full load—belt stretches and splices pull out. Slip injures belt surface.
25. Discharge chute choking—material catches on belt and tears it.
26. Idler pulleys broken—belt tears.
27. Lumps falling on return belt—jam under foot pulley and tear the belt.
28. Oil or grease dropping on belt—rubber softens and becomes loose.

TABLE 33.—ACTUAL LIFE OF CERTAIN RUBBER CONVEYOR BELTS

Case	Belt Width, Belt Plies, Belt Cover	Length of Conveyor, Center to Center, Feet	Speed, Feet per Minute	Material Carried	Tonnage According to "Rule"	Actual Tonnage	Length of Service	Belt Cost per Ton of Material Carried, Cent
1	42″×6, $\frac{1}{8}$″	359	325	1″ ore, 100 lbs. per cu. ft.	3,175,200	4,024,559	2 years, still running	
2	36″×5, $\frac{1}{16}$″	62	275	Screened coke, 30 lbs. per cu. ft.	401,760	238,237	14 months, worn out	.11
3	40″×8, $\frac{1}{8}$″	223	220	8″ stone, 100 lbs. per cu. ft.	1,784,000	2,061,132	3 years, worn out	.10
4	36″×8, $\frac{1}{8}$″	570	493	R. of M. coal, 50 lbs. per cu. ft.	3,693,600	2,808,350	$5\frac{1}{2}$ years, worn out	.13
5	36″×6, $\frac{1}{8}$″	265	595	R. of M. coal, 50 lbs. per cu. ft.	1,749,600	1,134,631	$2\frac{1}{2}$ years, worn out	.12
6	36″×6, $\frac{1}{8}$″	265	595	R. of M. coal, 50 lbs. per cu. ft.	1,749,600	1,084,640	2 years, worn out	.13
7	36″×7, $\frac{1}{8}$″	510	500	R. of M. coal, 50 lbs. per cu. ft.	3,304,800	1,461,520	3 years, worn out	.20
8	36″×6, $\frac{1}{8}$″	75	500	Large coke, 30 lbs. per cu. ft.	518,400	237,286	$3\frac{1}{2}$ years, worn out	.19
9	36″×5, $\frac{1}{16}$″	62	275	Small coke, 30 lbs. per cu. ft.	421,400	217,644	1 year, worn out	.12
10	36″×5, $\frac{1}{16}$″	25	400	Furnace coke, 30 lbs. per cu. ft.	168,480	522,205	$2\frac{1}{2}$ years, worn out	.03
11	36″×8, $\frac{1}{8}$″	165	....	Crushed ore, 125 lbs. per cu. ft.	1,069,200	7,313,400	38 months, worn out	.02
12	48″× 8, $\frac{1}{16}$″	612	400	R. of M. coal	7,050,240	1,339,352	41 months, still running	
13	48″× 8, $\frac{1}{16}$″	361	350	R. of M. coal	4,158,720	2,786,396	38 months, still running	
14	48″× 8, $\frac{1}{16}$″	150	350	R. of M. coal	1,728,000	3,172,915	53 months, still running	
15	42″× 8, $\frac{1}{16}$″	674	350	R. of M. coal	5,944,680	2,651,300	57 months, still running	
16	54″×11, $\frac{7}{32}$″	740	500	R. of M. coal	12,750,000	2,500,000	70 months, still running	

NOTE.—The belt in Case 11 was made of 42 ounce Duck.

The following actual cases in which complaint was made about the belt may be of interest:

1. A belt at a by-product plant lost a strip of its cover; the complaint was based on "poor quality of rubber." Examination showed that the direction of tear was *with* the travel of the belt, so that it could not have been done by anything catching on the moving belt. Further investigation proved that the belt had been torn while it was being put on the idlers. On the resistance of rubber to tearing, see page 30.

2. A belt gave only 180 days of service instead of a year as other belts had lasted. It had been cut by a tool or something sharp falling on the belt.

3. A belt in a cannery ran in a tin-lined trough with plows (see p. 158) at intervals to discharge the farmers' raw material into bins. The belt was injured by rough joints in the tin trough. When the material was dumped on the belt it contained nails, stones and sharp sticks which the plows forced into the belt.

4. A belt handling raw fruit in a cannery. On account of bad alignment, the edges were worn and the acid juice corroded the cotton fibers.

5. A belt carrying wet sand. Edges rubbed off, water got in, plies came apart.

6. A belt carrying crushed ore. Ends not cut square at the splice; belt ran crooked, edges rubbed off, plies came apart.

7. A cement company ordered a belt without stating that it was for hot material. The rubber dried out and plies came apart. Belts can be specially built to handle hot products in cement mills (see p. 23).

8. A belt was used with a tripper to discharge wet gravel into a row of bins. The tripper had no brush; the lower pulley forced sharp particles into the belt and the cover did not last long.

9. A coke belt lasted only one-third as long as was expected. The coke was not properly quenched; hot pieces burned holes in the belt.

10. A belt carrying crushed stone was reported to be wearing out rapidly. Investigation showed that it had been put on upside down with the $\frac{1}{8}$-inch rubber cover in contact with the pulleys; the side that carried the stone had only the usual $\frac{1}{32}$ inch of rubber intended for the pulley side.

11. When a certain case of excessive wear on one edge of a belt was investigated it was found that the gauge of a tripper track was too great for the wheels, the tripper pulled out of square with the belt and the latter bore hard against the side-guide rollers in the tripper frame.

12. To handle wet coal under unfavorable conditions it had been found economical to use balata belts. One of these lasted only a few weeks and was replaced on the claim that it was defective. It was discovered later that hydrochloric acid spilled on a floor above the conveyor had dripped down on the belt.

Comparison of Life of Belts.—The true measure of belt service is what the belt will do per dollar of investment or, in other words, what it costs per ton of material carried, or per year or month of service rendered. There is no standard of comparison for this because operating conditions in any

two plants are never exactly alike; what may be belt extravagance in one plant may be justifiable practice in another. A certain 36-inch 8-ply rubber belt in a Western smelter (see Table 33) carried over 7 million tons of ore in thirty-eight months at a cost of less than .02 cent per ton—a most excellent record. On the other hand, if a belt used in the throwing device shown in Figs. 167 and 192 lasts one week and carries 30,000 tons at a cost of .4 cent per ton, it is considered excellent service, although the rate per ton is very much higher.

It is often unfair to judge the quality or the suitability of various kinds of belts, rubber of different makes, stitched canvas, balata or solid-woven belts by a comparison of the life of one or two specimens. As has been stated, many belts must be replaced because of abuse or accident, not on account of wear in service. This fact is well known in the belt business; some concerns will sell belts on a guarantee of a certain life or tonnage "barring abuse or accident" with a fair certainty that under the operating conditions the belts will not have a chance to wear out, but will meet with one or more of the mishaps listed in Table 32, page 197. No belt, whatever the type or make, is proof against all of the items mentioned, and most of the mishaps will, if they occur, shorten the life of any belt.

Life of Rubber Belts in Service.—During the past thirty years concerns in the business of making and selling rubber belts have collected much valuable information about their life in handling materials of various kinds under many conditions. For reasons given above, this information cannot always be presented in a way to convince a prospective buyer that a certain kind of belt is the one best suited to his operating conditions. All that can be done is to say, with some authority, what has happened under similar conditions elsewhere. From this knowledge and some consideration of the principles given it is generally possible to make a fair guess as to which belt will give the best service in the particular place, per dollar of cost.

Table 33 prepared from information furnished by several manufacturers of rubber belts, gives, in addition to the comparison mentioned on page 196, a statement of the life and cost of certain belts per ton of material carried in plants carefully managed. Some of the costs per ton are exceptionally low; not one of them is high.

Belts protected from sun, snow and rain last longer than those that work in the open. At a large coke plant in the Middle West, rubber belts carrying run-of-mine coal on an open bridge spanning the storage pile average three years of service. Similar belts in the same system which are protected by enclosures average nearly five years.

Stitched Canvas Belts in Service.—Manufacturers of stitched canvas belts have furnished the following information:

Table 34 is a record of 5 conveyor belts used by one company in bulk cargo boats carrying crushed limestone and run-of-mine soft coal between ports on the Great Lakes. The belt formed part of a mechanical system for discharging the boat's cargo.

TABLE 34.—COMPARATIVE SERVICE OF BELTS CARRYING LUMP LIMESTONE AND R. OF M. COAL. BELT 48-INCH, 10-PLY, 175 FEET LONG

(Imperial Belting Co., Chicago)

Kind of Belt	Tons Carried	Cost of Belt	Belt Cost per Ton of Material Carried Cent
Rubber	416,249	$1287	.309
Solid-woven cotton	449,082	982	.219
Stitched canvas	574,402	1033	.180
Stitched canvas	895,731	1066	.119
Stitched canvas	1,227,480	1350	.110

At a sand and gravel washery plant in northern Illinois 300 feet of 30-inch 8-ply stitched canvas belt ran four seasons of eight months each and handled 1,497,000 tons of sand and gravel—a record for the plant.

At another plant of the same kind 400 feet of 24-inch 8-ply stitched canvas belt on an incline lasted six seasons of eight months each and handled 1,235,000 tons of sand and gravel. The long life of this belt may be attributed in part to the fact that it was not troughed but ran flat.

A stitched canvas belt, 18-inch 6-ply, 254 feet long, saturated with a Class 2 compound (see p. 48), lasted one year in an Illinois cement plant and carried 143,000 tons of hot cement, which was 4,000 tons more than the combined tonnage of two belts previously used. The canvas belt was replaced by another kind of belt under a guarantee of equal service, but the makers of it had to furnish four belts during the year to make good the guarantee. When the year was up, the cement company put in another canvas belt.

A 36-inch 6-ply canvas belt on an inclined conveyor, 430-foot centers, rising 80 feet, speed 450 feet per minute, over $22\frac{1}{2}°$ troughing idlers with snub drive from 54-inch head pulley, carried 1,400,000 tons of run-of-mine coal at a cost of less than $\frac{1}{5}$ cent per ton.

A 36-inch 8-ply canvas belt on 22° incline, 270-foot centers, running over 20° troughing idlers, 54-inch lagged head pulley, carried 1,000,000 tons limestone crushed to 6 inches and less.

A 24-inch 5-ply canvas belt, horizontal, carried 723,000 tons wet screened and washed coal for less than $\frac{1}{10}$ cent per ton.

CHAPTER XIII

WHEN TO USE BELT CONVEYORS

General Advantages of Belt Conveyors.—The range of uses of belt conveyors is very wide; they carry all kinds of bulk materials from clippings of tissue paper to ore in pieces weighing 200 to 300 pounds. Package goods of all sorts except the heaviest bales and barrels are successfully handled.

For many of these uses, the belt conveyor is the cheapest and best machine; for some it is the only machine. In some cases it is the only machine that will give the required capacity. Capacity is generally a matter of speed; in a belt conveyor, it is comparatively easy to get high capacity by using a speed high enough, for, so far as the driving of the belt is concerned, the possible speeds are far beyond all needs for conveying. Belts used in the transmission of power often travel 5000 feet per minute, but few belts except some used in grain conveying travel over 600 feet per minute. Chain conveyors, on the other hand, seldom travel over 250 feet per minute; at higher speeds, the shock and noise caused by chain links engaging with the sprocket wheels becomes objectionable and the wear in the chain joints becomes troublesome.

The machinery of a belt conveyor is usually simple in character and light in weight; it is not likely to break down without warning and it generally consumes less power for the work accomplished than any other form of conveyor.

The characteristics mentioned above have been responsible for the great development and wide use of belt conveyors. In the choice of a conveyor to do a certain work in a particular place they should be carefully considered, but at the same time some collateral disadvantages should not be overlooked. These do not apply to belt conveyors in general, but rather to specific uses as referred to below.

When to Use Belt Conveyors.—Since a belt conveyor will carry material horizontally or up an incline, or even do both in one machine, it will be of interest to discuss the merits of belt conveyors as compared with those of conveyors of other types and with combinations of elevators and conveyors.

Boiler Houses.—Distribution of coal in overhead bins can be accomplished by belt conveyor or flight conveyor. If the bin is a long continuous one, a belt with a traveling tripper will fill it, but if the bins are short, with spaces between, or separate round tanks, a flight convevor is more convenient for the discharge. Separate fixed trippers could be used, but they wear out the belt by repeated reverse bending and by throwing material

back on the belt when coal is to be carried past one or more of them. To move a traveling tripper over clearance spaces while the belt is empty and set it over various bins requires care and attention on the part of the attendant and may cause delays in the operation of the machine. Traveling trippers with large storage chutes to carry the discharge from the belt past clearance spaces and then drop it into the separated bins have been designed and patented, but are not in commercial use.

If the capacity required is less than 50 tons an hour, the work is less than what a 14-inch belt will do at a moderate speed. Twelve-inch belts are practically obsolete; 14-inch belts are hard to load with coal larger than 2-inch size. For small capacities a belt conveyor will cost more per foot run and will be burdened with the expense of the tripper. Narrow belts are apt to run crooked on standard commercial idlers and suffer damage when they are held to place by side-guide idlers.

If the capacity is 100 tons an hour or more, belts 20 inches or more in width can be used. These are easier to load with crushed coal and they run straighter than narrow belts. Belt conveyors in this class, as a rule, cost less than good flight conveyors and they consume less power.

If the conveyor is shorter than 75 or 100 feet a flight conveyor may be better. All of its length can be used for distribution; but in a belt conveyor, 15 or 20 feet is lost between the loading point and the point of first discharge to prevent the tripper from lifting the belt under the chute (see p. 169). A short belt wears out faster than a long belt (see p. 153), and since the terminals of a short conveyor use more power than the run of the belt, a belt conveyor in this class may not show any measurable saving of power as compared with a flight conveyor.

If the length of the conveyor is more than 200 feet, a belt will show a noticeable saving of power, and for lengths over 300 feet the expense of a belt conveyor is likely to be less than the cost of a double-strand conveyor with roller chains. A flight conveyor on a single chain will, however, cost somewhat less than the belt, but it will not handle large coal so well. It will require about twice as much power as a belt conveyor.

If quietness of operation is essential, use a belt. It must be said, however, that modern roller flight conveyors and double-strand roller-chain conveyors are much quieter than old-style flight conveyors.

If the conveyor is close up under the roof and receives from an elevator, a flight conveyor may fit in better. It requires less head room than a belt with a tripper; the loading chute can be shorter and can load the conveyor from the side. A belt should not be loaded from a side-delivery chute; to load it properly requires more height than to load a flight conveyor.

The belt is the costly item in a belt conveyor and at the same time the most vulnerable part. If it receives good care it will, in places suited to it, render service at a lower cost per ton of material carried than can be shown by any ordinary flight conveyor. But the belt may be damaged or ruined by a number of causes and then the charges for repair or renewal may be excessive. The ordinary causes of belt failure are given in Table 32.

A corresponding list of causes of failure of flight conveyors would be shorter. In general, a flight conveyor will stand more abuse and will work under bad conditions and under a lack of care that would be harmful to belts.

Transfer of Material without Distribution by Tripper.—In this case the belt makes a better showing than a flight conveyor except for short distances and low capacities. For feeders, a corrugated steel apron costs less than a wide belt with the right number of plies and the right thickness of cover. It is less likely to be injured by tools, sticks and hard, sharp lumps.

The belt will, however, make a cleaner delivery of material at the head end and will not spill so much on the return run.

On inclines, a flight conveyor will work at angles steeper than 25°, but a belt will not. For heavy tonnages a steep flight conveyor becomes costly and, as between the two, it may be better to use a longer belt at a flatter angle, or even two belts in series to raise the material to the required height. Transfer of coal at by-product plants is now done chiefly by belt. Here the distances are great and the capacities high. Flight conveyors would cost more to build and be more costly to drive. The disadvantages of belt conveyors in this work are the high first cost of the enclosing structures and the maintenance of them, and the fact that the bunkers and the loading points are spread apart over a stretch of ground which would otherwise be unnecessary. It is possible that such plants may be built more compactly in the future using large skip hoists or bucket elevators instead of belt-conveyors.

Crushed Ores, Gritty Materials.—Belts are in general use for this work.

Lump Rock, Sand, Gravel, Excavated Earth.—Belts do this work better than any metal conveyors, except that for very large rock, wide apron conveyors with wood or steel slats will stand abuse that would ruin belts. They cannot, however, be economically made as long as belt conveyors.

Ashes.—Many belts tried on this work have failed from damage by hot cinders. Moreover, if the ashes are handled wet, the cinders cut the cover and the water ruins the duck.

Coke.—Belts are in general use. The grit wears out metal conveyors too fast for economy. Coke is too friable to handle in scraper conveyors.

Wood Chips.—Belts are in general use. The work is light and does not require expensive belts.

Package Handling—Belts are in general use for all except the largest packages and the roughest work.

Zigzag Conveyors for Elevating Material.—In a few places a rising series of short inclined belts have been used to do the work for which a bucket elevator or a skip hoist would ordinarily be installed. They do not save in first cost; they take up more space and require more power than a single elevator. Some have been satisfactory to owners; in others the short belts wore out rapidly from slip due to the steep incline (see p. 142) and from abrasion at the loading point.

In the matter of spare parts to be kept on hand, there is this difference.

Repair parts for a flight conveyor can be kept on hand indefinitely, but a spare rubber belt should not be ordered too long before it is really needed, or it will deteriorate in storage (see p. 44). Canvas belts suffer less in this respect.

Comparisons.—Where comparisons have been made between belt conveyors and chain conveyors on the basis of tonnage, length or cost, the limits are not hard and fast and between them there are chances to exercise some personal preference. Both kinds of machines have been on the market for many years and have been to a great extent standardized. There is no "universal conveyor"; for handling heavy or gritty ores as in the Western smelters or in conveying heavy tonnages of coal over long distances, a flight conveyor is not generally considered, but in many places, boiler houses especially, it makes the best distributor even though it does take more power to run it. The older forms of flight conveyor were noisy, but modern roller flight conveyors, or roller chain conveyors are, by comparison, nearly noiseless; they are strong and rugged, easy to load, easy to discharge, spill less dirt outside, and they will work under bad conditions where a belt will fail. Some engineers have a prejudice against scraping coal in a trough. It does take more power, but it does not hurt the coal, even anthracite coal; and as for the wear on the trough the item of replacement is not an important one. It is a fact, however, that in plants like breakers in the anthracite coal region and tipples in bituminous districts where nearly all the coal of the country is prepared for market, chain conveyors are used far more than belt conveyors.

The right choice of a conveying machine depends upon:

1. A proper knowledge of the limits within which each kind works best. This implies some acquaintance with the disadvantages of each type of machine. The advantages are usually stated in manufacturers' catalogues, proposals and advertising, but the disadvantages are generally learned by experience.

2. A recognition of the fact that a machine highly successful in one place may be a failure in another place, even at the same kind of work.

3. Proper consideration of the auxiliaries necessary to the operation of the conveyor. A simple conveying medium with complicated auxiliaries for feeding or discharge may be less desirable than a more complex conveying medium with simpler auxiliaries. In a belt conveyor, for instance, the belt itself is simple and strong, but it requires the right kind of a chute to load it safely, and to effect a discharge at intermediate points, a tripper, with its many parts, may be necessary. A flight conveyor has a chain made up of many parts, but feeding is simple and discharge is merely dropping the material through a hole in the trough. A screw conveyor is not economical of power, but for low cost, simplicity of mounting, compactness, ease of feed and discharge, and for cleanliness and avoidance of dust, it is in many places preferable to any other kind of conveyor.

Long-distance Conveying by Belts.—On this subject, see page 107.

SECTION II.—BELT ELEVATORS

CHAPTER XIV

GENERAL DESCRIPTIONS

The Elements of a Belt Elevator.—A belt elevator for bulk materials consists of:

1. Buckets to contain the material.
2. A belt to carry the buckets and transmit the pull.
3. Means to drive the belt.
4. Accessories for loading the buckets or picking up the material, for receiving the discharged material, for maintaining belt tension and for enclosing and protecting the elevator.

Kinds of Belt Elevators.—Any kind of belt with buckets attached can

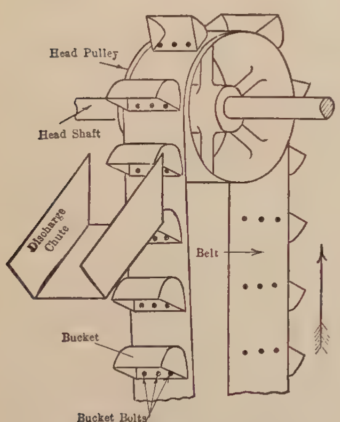


Fig. 200.—Head of Centrifugal Discharge Belt Elevator.

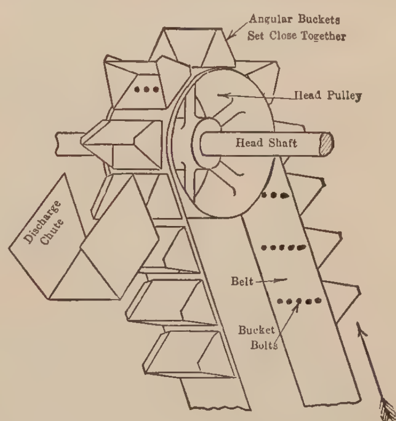


Fig. 201.—Head of Belt Elevator with Continuous Buckets.

be run around an upper pulley and a lower pulley, and it will elevate loose material. If the belt speed is high enough the contents of the buckets will be thrown out in passing over the upper pulley (head pulley) and will fall into a chute set to clear the descending buckets, some distance below the head shaft. This is a Centrifugal Discharge Elevator (Fig. 200); it may be

vertical or it may stand at an angle. Vertical elevators depend entirely on the action of centrifugal force to get the material into the discharge chute and must be run at speeds relatively high. Inclined elevators with buckets spaced apart or set close together may have the discharge chute set partly under the head pulley, and since they do not depend entirely on centrifugal force to put the material into the chute, the speeds may be relatively lower.

Nearly all centrifugal discharge elevators have spaced buckets with rounded bottoms; they pick up their load from a boot, a pit or a pile of material at the foot pulley.

If the buckets are triangular in cross-section and are set close on the belt with little or no clearance between them, the machine is a Continuous Bucket Elevator (Fig. 201). It can be used at high speed with centrifugal discharge as in some grain elevators, but this use is not common. The chief use of continuous bucket elevators is to carry difficult materials at slow speed. Discharge, in this case, is aided slightly by centrifugal force, the contents of each bucket pouring out over the inverted bottom of the bucket ahead of it, and into the head chute. The elevator may be vertical or inclined; to permit the buckets to be loaded directly from a chute, most elevators of this kind are inclined; very few pick up their load under the foot wheel.

Elevator Buckets.—The general requirements for an elevator bucket are as follows:

1. Dimensions large enough to pick up, hold, and discharge the largest pieces of the material handled by the elevator.

2. Cubic contents enough to give the required elevator capacity in pounds per minute, or tons per hour or per day, considering the speed of the belt, the bucket spacing, the regularity or irregularity of loading and the probably incomplete filling of the buckets.

3. Strength and stiffness to pick up its load without crushing or distortion.

4. Thickness of metal sufficient to resist wear to an economical degree.

5. Inside of bucket so shaped that material will not stick there and fail to discharge.

To meet these requirements and to handle the many kinds of bulk materials, buckets for belt elevators are made in several styles:

1. Buckets with rounded bottoms, the kind generally used in centrifugal discharge elevators.

2. Buckets with angular bottoms, sometimes used in high-speed centrifugal discharge elevators for grain, but more often employed in continuous bucket elevators for coarse, heavy materials.

With respect to their construction and the material of which they are made, elevator buckets may be classified thus:

1. One-piece or three-piece buckets of tin plate or light-gauge sheet steel, with seamed corners and reinforced by steel bands. These are used for flour-mill products and for grain.

2. One-piece buckets of heavier steel, pressed to form and riveted or

welded, without reinforcing bands. Used for grain and for materials heavier than grain.

3. Hot-pressed or cold-pressed seamless sheet-steel buckets, used for grain and other bulk materials not too heavy.

4. Cast malleable-iron buckets, for coal, ores, minerals and other coarse, heavy materials.

5. Two-piece or three-piece buckets of heavy steel plate made with angular bottoms for continuous bucket elevators.

Elevator buckets are described more fully in Chapter XVI.

Elevator Belts.—The general requirements for an elevator belt are:

1. Sufficient flexibility to wrap easily around the head and foot pulleys.

2. Width enough to fasten the elevator buckets securely and to avoid twisting or turning over on the ascent.

3. Thickness sufficient to transmit the working pull without excessive stretch, to back up the buckets without deflection and to resist the tendency of the bucket bolts to pull through the belt.

4. A protective cover or a body of fabric thick enough and strong enough to resist, to an economical degree, the surface wear in elevators that handle sharp, abrasive materials.

Practically all elevator belts in this country are made of cotton fiber in some form; they are:

1. Rubber belts.
2. Stitched canvas belts.
3. Balata belts.
4. Solid-woven belts.

These are described briefly in Chapter I and more fully in Chapters III and XVIII. Leather makes a good elevator belt for dry work; it was in general use for that purpose up to fifty years ago, but now fabric belts are cheaper, more economical, and better suited to most cases. Elevators with woven-mesh steel-wire belts (Fig. 62) have been used in Europe for light service, but they are unknown in this country. Elevators with buckets fastened to two or more parallel strands of wire rope have been tried at various times but without success.

Driving the Belt.—The elevator belt is driven by the frictional contact between it and the rim of the head pulley, and since it is not possible to use a binder pulley or snub pulley against the bucket side of the belt, the angle of contact is limited to about 180°. The ability of the head pulley to drive the belt depends (see p. 109) on the angle of belt wrap and the coefficient of friction between the belt and the pulley rim; in an elevator both of these are fixed within certain limits, and it is not so easy to increase the driving effect as in a belt conveyor (see p. 110) where the angle of wrap can be made larger than 180°. To get more pull in an elevator it is necessary to put tension on the belt, relatively more than in conveyors, and this leads to the use of higher unit stresses in elevator belts than in conveyor belts.

Belt elevators have been driven at the foot, but the drive is always uncertain and often troublesome (see p. 283).

Accessories for Loading the Buckets.—In some forms of elevating and conveying apparatus it is possible to deposit separate charges or loads in consecutive buckets as they pass, by means of a mechanical loader, but at the speeds at which centrifugal discharge elevators run that cannot be done; the material cannot be guided into a bucket moving at a rate of 3 to 10 feet per second, and the impact would scatter and spill it. At the lower speeds of continuous bucket elevators, 80 to 200 feet per minute, the difficulties of mechanical loading are less, but still serious enough to make this process expensive and troublesome. It is much easier to load continuous buckets by means of a chute, especially when the elevator is inclined, so that what one bucket misses, the next one will catch. In centrifugal discharge elevators, however, it is never possible to load buckets from a chute without spill, and it is not often attempted; it is necessary in all cases to let the buckets pick up some or all of their load as they pass around the foot wheel or as they enter the vertical run. If the elevator digs from a pit or a pile, the material is naturally confined to the path through which the buckets sweep, but otherwise a box or boot is used to form a mounting for the foot shaft and keep the material within reach of the buckets.

Belt Tension.—In most cases the foot-shaft bearings are adjustable in position either as take-up bearings separate from the boot or as sliding bearings which form part of the boot. Sometimes the foot-shaft bearings are fixed, then the take-up bearings are placed at the head of the elevator.

Discharge at the Head.—In some forms of chain elevators the buckets discharge on the lift, but all belt elevators discharge at the head into a chute set to catch the material, either as it is thrown out by a centrifugal discharge elevator (Fig. 200) or as poured out by a continuous bucket elevator (Fig. 201). The position of the chute and the discharge of material from the buckets depend upon three factors:

1. The speed of the belt.
2. The diameter of the head pulley.
3. The spacing and shape of the buckets.

At the same time, the loading or pick-up at the foot depends upon:

1. The speed of the belt.
2. The diameter of the foot pulley.
3. The spacing and shape of the buckets.

The best speeds for the pick-up and discharge of different materials have been determined by trial and experiment, and have been confirmed by years of successful practice. They agree so well with results given by analysis that it will be of interest to show how they can be established by some consideration of the theory of the subject. This discussion will at the same time serve as an introduction to the further consideration of the design and construction of belt elevators.

CHAPTER XV

CENTRIFUGAL DISCHARGE ELEVATORS

Pick-up and Discharge of Elevator Buckets.—When a mass of material of weight W is passing around a wheel it is under the influence of two forces: one, gravity, acts vertically downward with a force W; the other, centrifugal force, acts radially outward from the center of rotation with a force $= \frac{Wv^2}{gR}$, where v = velocity of the mass in feet per second, g = acceleration of gravity = 32.2 feet per second, and R is the radius in feet to the center of rotation.

The action of buckets passing under a foot wheel or over a head wheel is shown in Fig. 202. In the position 3, centrifugal force acts horizontally outward, gravity acts downward, and the diagonal resultant obtained by completing the parallelogram of forces shows by its *direction* that the resultant pressure is downward within the bucket, and by its *length* on the scale to which the other forces are drawn, that the pressure is $\sqrt{2} = 1.414$ times the weight of the mass in the bucket if centrifugal force and weight are equal in amount. At 4, the pressure decreases; at 5 it becomes zero if the two forces are equal; and at 6 it acts to propel the mass from the bucket toward a chute set to catch the material and with a force which on the scale of the drawing equals about three-fourths of the force of gravity.

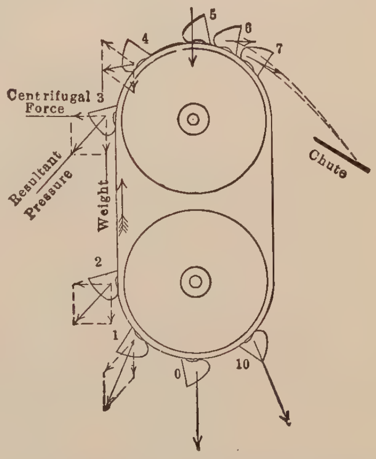


Fig. 202.—Action of Forces on Pick-up and Discharge. Centrifugal Discharge Elevators at Speeds Given in Table 35.

Best Speed for Elevator Discharge.—In order to deliver material to the chute without spilling or scattering, it would seem that at a place near the top of the wheel the two forces, weight and centrifugal force, should be equal in amount, because then, for the position 5, the mass within the bucket will be in equilibrium or a state of suspension, neither tending to fly out upward nor to fall out on the wheel, but ready to move out freely when the resultant of the two forces on the descending side of the wheel

urges the material toward the mouth of the bucket. That condition of equilibrium exists when

$$W = \frac{Wv^2}{gR} \text{ (see above) or } v^2 = gR. \quad \ldots \ldots \quad (1)$$

or since

$$v = \frac{2\pi R N}{60}$$

where N = number of revolutions per minute, then

$$N = 54.19 \frac{1}{\sqrt{R}}. \quad \ldots \ldots \ldots \ldots \quad (2)$$

This relation between radius of the head wheel and its revolutions per minute holds good in practice for the discharge of liquids or dry free-flowing materials like grain from elevator buckets of ordinary shape.

Table 35, calculated from equation (2), shows diameters of head wheels and corresponding speeds for the entire range of sizes used in centrifugal discharge elevators. In the calculations, assumptions have been made for the thickness of belt and the projection of buckets ordinarily used with each size of head wheel, and R has been taken as the distance from the center of the headshaft to the center of gravity of an average load in the bucket.

TABLE 35.—HEAD WHEELS AND SPEEDS, CENTRIFUGAL DISCHARGE AT HIGH SPEED

Diameter of Wheel, Inches	Revolutions per Minute	Belt Speed Feet per Minute	Diameter of Wheel, Inches	Revolutions per Minute	Belt Speed, Feet per Minute
12	69	217	42	39	429
15	62	247	48	37	465
18	56	264	54	35	495
21	53	292	60	33	518
24	50	314	66	32	553
27	47	333	72	31	584
30	45	353	84	29	637
33	43	372	96	27	679
36	41	386	108	25	707
39	40	408	120	24	754

In this discussion, a centrifugal discharge elevator is one that runs at a speed high enough to discharge the contents of the buckets clear of a vertical run of descending buckets.

The parabolas drawn from positions 6 and 7 (Fig. 202) represent the path of discharge from buckets in those positions. The material all clears the head wheel and will enter a chute with its upper end placed to clear the descending buckets and about 10° of arc below the center of the wheel.

The figure shows successive positions of one bucket, not simultaneous positions of adjacent buckets.

The upper half of Fig. 202 represents the action on a head wheel and the lower half shows what happens on the bottom of a foot wheel. If the material were fed in on the descending side of the foot wheel, a bucket at position 10 could not retain it because the resultant is directed toward the mouth of the bucket. At position 0 the resultant falls within the bucket, but nearly all the material would flow out because its surface would tend to lie at right angles to the line of pressure, just as in a vessel of water whirled around an axis, the water surface tends to stand at right angles to the line of pressure exerted along the radius of rotation. At 1 the tendency to squeeze material out of the bucket is less than at 0, and at 2 it is still less; but even there it would be impossible for a bucket to retain a full load under the influence of a pressure which is 1.4 times the force of gravity and is directed at such an angle that when the material did shift to square itself with the line of pressure some of it would fly out over the front lip of the bucket.

These considerations show that it is impossible for buckets to carry free-flowing material like grain around or from under a foot wheel when the speed of travel is that given in Table 35. But if the boot is so arranged that the material is fed in above the level of the foot shaft, as in Fig. 207, then the bucket is loaded on a straight lift above position 2, where centrifugal force no longer acts on it. When a boot is fed at the back, the material is swept around by the buckets, but the buckets fill at the front (see Fig. 209).

Effect of Higher Speeds.—Fig. 203 shows the effect of making centrifugal force equal to twice the force of gravity, a condition which exists when for a wheel of given size the revolutions per minute $= 1.414 = \sqrt{2}$ times the values given in Table 35. Considering first the upper half of the diagram, as representing the top of the head wheel, we see that the contents of a bucket rising to the position 3 will be acted upon by a force which is $\sqrt{5} = 2.23$ times as great as the force of gravity. The direction of the force shows that some of the material must be suddenly spilled over the front lip of the bucket. Once over the lip, the spill will fly outward and fall down the rising leg of the elevator. The resultant pressure decreases at 4, but at 5 the direction of the pressure is vertically upward and some of the contents of the bucket would be thrown in that direction. If the scale of the drawing represents a 96-inch wheel making 38 revolutions, the spill would rise about 3 feet and then fall down. Any material remain-

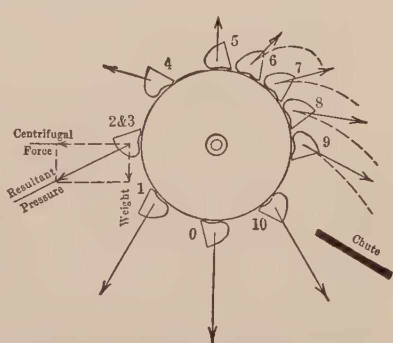


Fig. 203.—Action of Forces on Pick-up and Discharge. Centrifugal Discharge Elevators at Speeds 41 Per Cent Higher than Table 35.

ing in the buckets at 6, 7, 8 and 9 would pass off in the parabolic curves shown in the figure.

It is evident that with the spill beginning at 3 and continuing past 6 there would be a whirling shower of material all around the wheel and that only a part of what the buckets picked up at the foot would reach a chute placed in any position near the head wheel.

Since the direction of the resultant of pressure shows the direction in which the material starts to leave the bucket and since for positions 5, 6, 7 and 8 (Fig. 203) these lines do not point directly toward the mouth of the bucket, there is a possibility that some of the material will be trapped within the bucket and descend with it, or at least require the chute to be set low to catch the tail end of the scattering discharge.

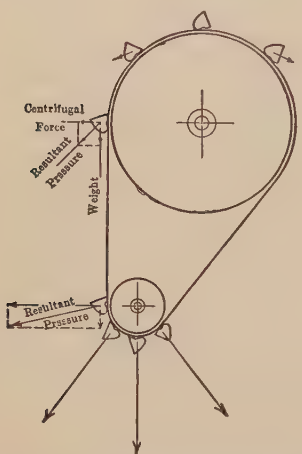


FIG. 204.—Pressures Affecting Pick-up in a Grain Elevator with Small Foot Pulley.

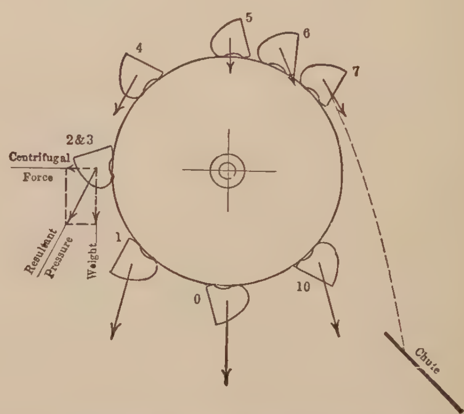


FIG. 205.—Pick-up and Discharge at Speeds 30 Per Cent Lower than Table 35.

When the lower half of Fig. 203 is compared with the lower part of Fig. 202 it appears that the pressures which would force material out of the buckets under the foot wheel are greater in amount and even less favorable in direction at the higher speeds. The lower half of Fig. 203 would represent the action of forces in an elevator boot where the foot pulley is half as large as the head wheel, but if the pulley is only one-third or one-fourth as large as is usual in large grain elevators the centrifugal forces are then three or four times as great at the foot as at the head since the force varies as $\frac{1}{R}$ for a constant value of v (belt speed). Such conditions are even more unfavorable for filling the buckets unless the loading is done after the buckets are on the straight vertical lift, as in Fig. 207.

Fig. 204 shows, on a smaller scale, a grain elevator with a 96-inch head

wheel and a 24-inch foot wheel. The resultant pressures (shown to proper scale) which prevent the grain from entering the buckets while they are in contact with the foot wheel are 6 or 7 times as great as those which throw the grain out of the buckets on the head wheel. It is quite evident that the buckets can take no load below the level of the foot shaft, but must be filled after they are on the vertical run.

Effect of Lower Speeds.—Fig. 205 shows the effect on pick-up and discharge when the speed is lower than the best speed in the ratio 1 to $\frac{1}{\sqrt{2}}=1$ to .7, a condition which makes centrifugal force one-half the force of gravity and N about seven-tenths the values given in Table 35. Considering the upper half as representing the top of the head wheel, a loaded bucket rising to the position 3 will not spill any of its load because the resultant pressure differs very little from W (due to the force of gravity) either in amount or in direction. The same condition exists at 4, but at 5 there is a downward force equal to $\frac{W}{2}$ which would tend to spill some material out of the bucket onto the top of the wheel. At 6 the resultant shows that more would spill out and at 7 the remaining material would be thrown clear of the wheel and could be caught in a chute, the upper end of which is set to clear the descending buckets and on about 45° of arc below the center of the wheel.

Such a discharge can be used for an inclined elevator, but it is too slow for a vertical elevator.

For a discussion of the discharge from inclined elevators, and of the point at which discharge begins see Chapter XXII and Fig. 285.

Belt Speeds for Different Materials.—From what has been said it is evident that the figures of Table 35 apply to free-flowing materials which can be piled deep in a boot and to buckets which finish their loading while on the vertical run above the center of the foot wheel.

It does not apply to substances like coal, ores, minerals, cinders, etc., which are not free-flowing, or which contain lumps, or which are apt to be wet and stick to the buckets. It is not safe to pile such materials deep in a boot; the work of pulling buckets through the mass would be wasteful of power, and buckets would be broken or torn loose by the severe strain. When the materials of that kind are handled in a centrifugal discharge elevator with spaced buckets, the feed must be tangent to the sweep of the buckets and not so high up that the foot wheel would be dangerously buried in case the amount of feed should exceed the lifting capacity of the buckets for a short time. It is customary to use sloped front boots (Fig. 259) for such work and the travel of the buckets *while on the foot wheel* must be limited to a speed at which centrifugal force will not throw material out of the bucket at positions 1 and 2 (Fig. 202). This tendency to throw material out is resisted at position 1 by the pressure between the moving bucket and the material flowing into the boot or lying on the bottom of the boot; and at position 2 by the fact that coarse substances like coal,

minerals, etc., are not so free-flowing as to be squeezed out of the bucket by a resultant pressure of the direction and intensity shown in Fig. 202.

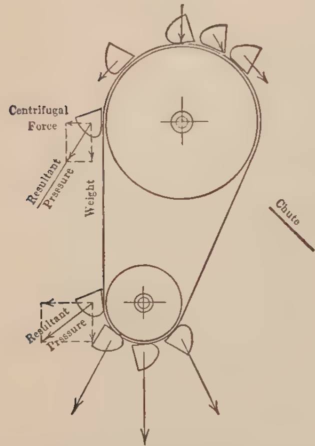


Fig. 206.—Pick-up and Discharge at Speeds given in Table 36. Foot Wheel Half as Large as Head Wheel.

Nevertheless, it is not practicable to use the speeds of Table 35 for coal, ores, minerals, cinders and similar coarse materials, and there are other reasons why that table should not be used for dusty materials. For these various materials, and even for free-flowing materials under certain conditions it is proper to use the speeds of Table 36 which are 82 per cent of those of Table 35. At these speeds centrifugal force is two-thirds the force of gravity and the conditions of discharge are shown in the upper part of Fig. 206.

Reasons for Table 36.—Some important reasons are given in a paragraph above; to these may be added the following:

1. Foot wheels for various reasons are often made smaller than head wheels. Centrifugal force is then greater than at the head, since for the same belt speed or velocity v, centrifugal force varies as $\frac{1}{R}$ (see p. 211); that is, it is twice as great for a wheel half as large. That difference, in the greatest degree existing in practice, is shown in Fig. 204, but even where the belt speed is less and the difference between the foot wheel and the head wheel is not so great, the action at the foot may be such as to prevent buckets from loading properly. Fig. 206 shows to scale what happens in an elevator for coal, etc., where the speed is according to Table 36 and the foot wheel half as large as the head, say, 18 and 36 inches, respectively. The pressures due to centrifugal force which hinder the filling of

TABLE 36.—HEAD WHEELS AND SPEEDS, CENTRIFUGAL DISCHARGE AT MODERATE SPEED

Diameter of Wheel, Inches	Revolutions per Minute	Belt Speed Feet per Minute	Diameter of Wheel, Inches	Revolutions per Minute	Belt Speed, Feet per Minute
12	56	176	39	33	337
15	51	200	42	32	352
18	46	217	48	30	377
21	43	237	54	28	396
24	41	257	60	27	424
27	39	276	66	26	449
30	37	290	72	25	471
33	36	311	84	23	506
36	34	320	96	22	554

the buckets while they are on the 18-inch foot wheel are more than three times as great as those which throw the material out of the buckets while they are on the 36-inch head wheel. If the foot wheel were 24 inches in diameter (ratio $\frac{2}{3}$), the pressures as to intensity and direction would be the same as those shown in Fig. 202; if the wheel were 27 inches (ratio $\frac{3}{4}$) they would be more favorable for the filling of the buckets; if the wheels were of the same size, i.e., 36 inches, the opposing pressures would be still less, the buckets would take their load lower down in the boot, with less turmoil and stir in the material, and they would carry, in most cases, a fuller load.

2. If the materials are picked up and discharged at high speed, the wear on buckets is serious; there is the risk of damaging them or tearing them loose, the strain in the elevator belt or chain is apt to be injurious, the wear on the head chute may be objectionable, and with some friable materials, like coal, the breakage from striking the chute violently may be a disadvantage.

3. Overcoming the friction losses at high speeds means a waste of power.

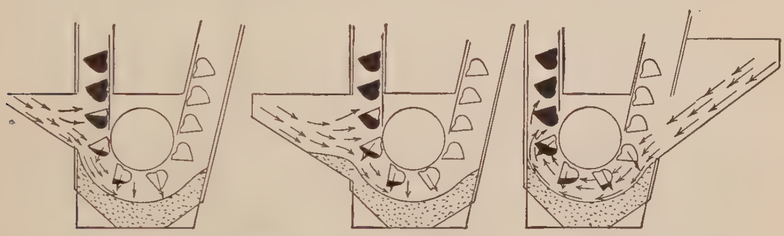


Fig. 207. Fig. 208. Fig. 209.

Fig. 207.—Pick-up in High-speed Grain Elevator, with Front Feed at Level of Foot Shaft.
Fig. 208.—Front Feed below Level of Foot Shaft.
Fig. 209.—Back Feed.

Practical considerations like these have established the following rules:

High Speed Grain Elevators.—If the material is dry, granular, free-flowing, not abrasive, not damaged by picking up or throwing down at high speeds, and if high capacity is desired, use Table 35.

This is preeminently the table for high-speed, high-capacity grain elevators.

Boots for high-speed grain elevators should have the front feed inlet so high and the front of the boot and the lower part of the casing so shaped that the buckets can complete their loading after passing above the center of the foot wheel in its upper position. Fig. 207 shows how grain is picked up in such a boot; if the feed in front is low, as in Fig. 208, the pick-up is not improved, the buckets may not take a full load, some of the grain may lie comparatively dead in the feed hopper, or the chute may be choked. If the feed is at the back (Fig. 209) there is no gain in placing the chute high, but if it is tangent to the sweep of the buckets in the upper position of the foot wheel, then if the elevator is choked to a standstill before the

feed is stopped, less grain will flow into the boot and less will have to be shoveled out to clear the boot in starting up again.

Discharge chutes for high-speed grain elevators may be set so that the lip just clears the buckets and lies on an arc 15° or 20° below the center of the head wheel. On this point, see page 303.

Limitations of High Speed.—So far as discharge is concerned, Table 35 applies to all kinds of centrifugal discharge grain elevators, but there are certain practical limitations to its use: when the feed chute is high and the buckets finish their loading above the level of the boot wheel there is danger of burying the wheel if the feed should be too heavy, or if the driving motor should stop by failure of electric current, or if the elevator should slow down or stop for any other reason. This is particularly true when the wheel is all the way down in the lowest position of take-up travel. High-speed grain elevators with foot wheels one-fourth as large as the head wheels are more liable to choke than those with foot wheels relatively larger, or where the speed is more moderate. Hence, unless the elevator belt is relatively wide in proportion to the projection of the buckets, so that it has plenty of pulley contact at the head, and unless plenty of power is provided to pull the buckets through an accidental accumulation of material in the boot, it is better to use a moderate speed, like those given in Table 36, and make the foot wheel at least half as large as the head wheel.

A moderate-speed grain elevator with a foot wheel half as large as the head wheel can use any of the standard makes of boot with rounded bottom and in which the bottom of the feed chute enters at or above the level of the foot shaft in its lowest position. If the foot wheel is not so large as the ratio $\frac{1}{2}$, more power is spent in stirring up the grain in the boot and unless the feed chute is set relatively higher up, the buckets will not fill properly and the feed chute may choke.

Discharge chutes for grain elevators at the speeds of Table 36 should be set about 30° below the center of the head shaft.

If the material is fine, dry and dusty the speed should be kept low to give the buckets time to free themselves of air in passing through the material in the boot; otherwise they do not fill properly but stir up the material uselessly and raise an objectionable dust. Besides that, the fan action of the pulley and buckets at the head may disturb the air enough to blow the material out of the chute or waste it down the elevator legs.

For flour, bran, chaff and similar mill products of light weight use Table 36 up to 24-inch head wheels, but not beyond. The foot wheel should be as large as the head wheel if possible, certainly not less than three-fourths of that size. The feed should be rather high; the lip of the discharge chute should be 20° to 25° below the center of the head wheel.

For fine cement, pulverized lime and other dry, dusty substances weighing more than 50 pounds per cubic foot, use Table 36 up to 42-inch head wheels, but not beyond. The ratio of head-wheel to foot-wheel diameter should not be over 4 to 3; if for any reason the diameter of the foot wheel is limited to 18 inches, for example, the head wheel must be in proportion;

in this instance it should not be larger than 24 inches. The feed should be rather high; the lip of the discharge chute should be about 30° below the center of the head shaft.

If the material is hard, gritty and lumpy, like coal, ashes, coke, stone, ores, salt, fertilizers, coarse chemicals, or if it is moist at times, as coal is, and apt to stick to the buckets, use Table 36. The speed of bucket travel and the number of revolutions of the head shaft are determined by the allowable wear and tear on the buckets and on the belt or chain, and not especially by the discharge of the buckets at the head. Belt elevators carrying coal crushed to $1\frac{1}{2}$ inches and less are run successfully at nearly 500 feet per minute on 72-inch head wheels; but 400 feet per minute (54-inch wheels) may be considered the permissible limit for belts carrying coal larger than 2 inches and a still lower speed, say, 350 feet per minute (42-inch wheels), should be used for rougher, harder and more abrasive materials to keep wear and tear within tolerable limits. Chain elevators are never run over 350 feet per minute, and on very coarse materials 300 feet is a safer limit. If the lumps are larger than 3 inches it is better not to use a centrifugal discharge elevator, but rather a slow-speed machine of some other type.

For ease in pick-up, and to fill the buckets properly, make the foot wheel at least two-thirds as large as the head wheel. If the space at the foot is limited, and must be made no larger than, say, 18 inches, then the head wheel should not be larger than 27 inches. Fig. 206 shows that when the head wheel is twice the size of the foot wheel the pressures which hinder the filling of the buckets are more than three times the forces which throw materials out of the buckets at the head. Fig. 202 shows how the pick-up is improved when both wheels are of the same size. In general, the larger the foot wheel, the easier the pick-up and the better the filling of the buckets. On this point see p. 214.

If for a certain capacity it is necessary to run buckets at, say, 350 feet per minute, the head wheel should be 42 inches in diameter (see Table 36) and the foot wheel should be at least 28 inches, 30 inches would be better, 36 inches still better. If there is room for only a 20-inch foot wheel, for instance, there are several things which may be done: (1) make the head wheel 20 inches $\times\frac{3}{2}=30$ inches, run the belt at 290 feet per minute (Table 36) and get the required capacity by a closer spacing of buckets or by using larger buckets; (2) make the head wheel larger than 30 inches and the belt speed more than 290 feet, and have an elevator wasteful of power, and with a capacity not up to normal because of buckets running only partly full; (3) use some other kind of elevator.

The boot for coal and similar coarse substances may be of any shape that will deliver the material to the sweep of the buckets if the following requirements are met: (1) it must not choke the feed inlet with dead material when the foot wheel is in its high position with the take-up all the way up; (2) the feed must not be so high that it can easily overflow or swamp the foot wheel when it is in its lowest position. For these reasons, most grain

boots do not make good boots for coal, etc. The boots generally sold for this service are made with sloped fronts (see Figs. 259 and 261).

The discharge chute should be set with its lip at least 30° below the center of the head shaft; 45° is better for coal and other materials which are damp at times, and if the material is wet and fine it is well to place the chute even lower than the 45° line.

Table 37 refers to elevators in which the discharge occurs without the assistance of centrifugal force. These are generally slow-speed inclined elevators, for which see Chapter XXII.

TABLE 37.—HEAD WHEELS, SPEEDS, BUCKET SPACING, INCLINED ELEVATORS, CENTRIFUGAL DISCHARGE AT LOW SPEED

Diameter of Wheel, Inches	Revolutions per Minute	Belt Speed, Feet per Minute	Bucket Projection, Inches	Bucket Spacing, Inches	Diameter of Wheel, Inches	Revolutions per Minute	Belt Speed, Feet per Minute	Bucket Projection, Inches	Bucket Spacing, Inches
12	35	110	3	9.4	27	24	175	7	21.2
15	31	124	$3\frac{1}{2}$	11.8	30	23	180	8	23.5
18	28	132	4	14.1	33	22	190	9	25.9
21	27	147	5	16.5	36	21	198	10	28.2
24	25	157	6	18.8					

Elevators for liquids can be run at the speeds of Table 35 with buckets of the shape shown in Fig. 202. In order to be filled full, the buckets should complete their loading above the center of the foot shaft; or in other words, the foot shaft must be submerged. If for any reason the foot shaft must be kept above the level of the liquid in the tank, or boot, it is important to make the wheel large, first, to maintain a good depth of liquid below the shaft for the buckets to act on; second, to avoid the high pressures, almost radial in direction (see Figs. 204 and 206) which prevent the buckets from filling under the foot wheel when the latter is smaller than the head wheel.

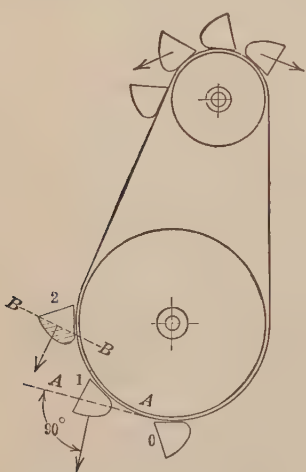


Fig. 210.—Pick-up of Liquids in Centrifugal Discharge Elevator.

If the elevator has buckets of ordinary shape it cannot be expected to carry a volume of liquid per minute or per hour based upon the rated liquid capacity of the bucket, as stated in makers' catalogues, because even if the foot wheel is larger than the head wheel, the directions of the resultant pressures under the foot wheel are such as to squeeze some of the liquid out of them. Buckets with a high front like Fig. 226 are better in that respect. Fig. 210 shows the pressures in an elevator run at a speed given in Table 35 and with a foot wheel twice as large as the head wheel.

Although elevators are not built with such large foot wheels, the figure is useful in showing how the filling can be improved in a high-speed elevator. Since the surface of the liquid in the bucket tends to lie at right angles to the line of resultant pressure (see p. 213), the contents of a bucket at position 1 would lie parallel to $A—A$; and at 2, $B—B$ represents the surface, at that instant, of what the bucket holds. Of course, the pressure due to the depth of liquid exerts a modifying influence at position 1, and the splash or the disturbed level of liquid in the boot may put more in the bucket at position 2; nevertheless it is an observed fact that if the buckets of a centrifugal discharge elevator are expected to pick up liquid below the level of the foot shaft they lift only a small fraction of their nominal capacity.

Pulps handled in the wet concentration of ores, and other similar flowing mixtures of solids and liquids, behave like liquids as far as pick-up is concerned; and what is said above applies to them also, that is, for a good pick-up the foot pulley should be larger than the head pulley. At the same time, the head pulley must not be too small. Table 35 gives proper relations of belt speed and diameter of head pulley for high speeds. Table 36 gives similar data for slower speeds.

If the percentage of solids in the pulp is high, and if the solids are heavy and tend to settle in the bottom of the bucket on the lift, the contents of a bucket are not discharged in a single mass, but the bulk of the water leaves first and the solids are later in leaving. This has the effect of delaying the discharge, and hence while the head receiver or chute for liquids like water or very thin pulps may be set quite high at the head of a centrifugal discharge elevator, say, 15° or 25° below the center of the head shaft, the lip of the receiver should be considerably lower for heavy pulps in which the percentage of solid is high. For these 45° is often not enough; 10° or 15° additional may be necessary in some cases.

If the receiver is set low for the reason mentioned above, it is better to run the elevator at a speed given in Table 36; the pick-up at the lower speed will be somewhat improved and the buckets will have a chance to take a larger load.

Delayed Discharge.—The theory of elevator discharge discussed in the preceding paragraphs assumes that the material leaves the bucket with no frictional resistance. It is possible to assume a coefficient of friction between the material of which the bucket is made and the contents of the bucket, and from a known velocity and radius of rotation, calculate how far beyond the vertical a bucket will be before the tangential force overcomes the frictional resistance within the bucket. Such calculations lead to no practical result. The coefficient of friction is an uncertain quantity and in comparison with roughnesses and dents in the buckets and the resistance offered by one or two rows of nuts and washers across the discharge opening its effect is small.

All that can be said with certainty is that, owing to the items mentioned, some of the material is delayed in discharge and some may be spilled. When the head chute of an elevator is set according to the rules given above

it will catch practically all of the material, including that which is delayed in discharge. In any elevator there is always some scatter or spill which will not reach the chute, but that amount should be small and it cannot be avoided.

Rules for Speeds of Centrifugal Discharge Elevators sometimes state the travel of the chain or belt in feet per minute without reference to the size of the head wheel. *Such rules are worthless.* For proper discharge, each size of head wheel has its own best speed. Those given in Table 35 and Table 36 represent, respectively, the best speeds for free-flowing materials and also for those which are heavy, coarse and not free-flowing. The figures for the number of revolutions per minute are whole numbers which differ from results of equation (2), page 212, by less than unity. Where it is not convenient to get these exact speeds by the use of standard sizes of pulleys, gears, sheaves, etc., used in the transmission of power to the elevator head, a revolution more or less will make no noticeable difference in the pick-up or in the discharge.

For emphasis it may be worthwhile to repeat the statement that the success of a centrifugal discharge elevator may depend on the size of the foot wheel and the way material is fed to the buckets. If the foot wheel is smaller than the head wheel it may revolve so fast that the buckets pick up no material at all while they are passing around the wheel, especially if the material is fine and free-flowing. With such materials the feed should not be too low. When elevators are equipped with low-feed boots and small foot wheels, they may work, but at a reduced capacity. Sometimes they are absolute failures.

The fact that grain buckets do not take their load below or behind the foot pulleys is quite evident when a boot like that shown in Fig. 271, page 295, is fed at the back, where the buckets are going down. At such times, unless the front opening is closed, not only will dust come out, but the grain itself will be thrown clear out of the boot. This is especially true when, as in many grain elevators, the bottom of the front opening is lower than the upper position of the foot shaft. For the best feed from the front, the opening should be about as shown in Fig. 271.

Evidence from Photographs.—The speeds and revolutions of head wheels given in Tables 35 and 36 are confirmed by successful practice and have been verified by instantaneous photographs of the discharge from the buckets. Some of these are given in Figs. 211, 212, 213, 214, which show the action of 8 by 5 inch buckets spaced 12 inches apart and running over 36-inch head wheels at various speeds. In Fig. 211 the wheel makes 35 r.p.m. and the buckets make a clean discharge of pebbles weighing 80 pounds per cubic foot. In Fig. 212 there is a clean discharge of oats at the same speed, although oats weigh only 30 pounds per cubic foot. In Fig. 213 the wheel makes 20 revolutions or 180 feet per minute chain speed, but the pebbles from bucket A are beginning to spill out on the chain while the discharge from B hits the inverted bottom of C. Fig. 214 shows a bad discharge of oats at 23 r.p.m. = 210 feet per minute.

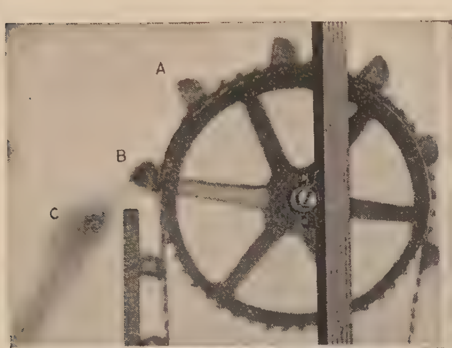


Fig. 211.

Fig. 212.

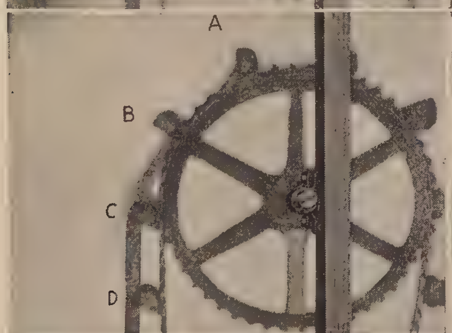


Fig. 213.

Fig. 214.

Figs. 211–214.—Instantaneous Photographs of Centrifugal Discharge at Various Speeds of 36-inch Head Wheel.

Point where Discharge Begins.—In the calculations for Table 35 it is assumed that centrifugal force equals the force of gravity, or $v^2 = gR$ (Equation 1, p. 212) and then discharge begins at the top of the wheel; but the lower values of Table 36 are based on centrifugal force $= \frac{2}{3}$ the force of gravity, or $v^2 = \frac{2}{3}gR$, so that the values of v in Table 36 are $\sqrt{.66} = .82$ of those in Table 35. For this condition discharge will begin later and at angle from the vertical whose cosine is $\frac{2}{3}$ (see p. 312). This angle is about 48°. The correctness of this reasoning is shown by Fig. 211, where the contents of bucket A, about 45° from the vertical, are just beginning to come out, the mass of pebbles at C being entirely clear of bucket B. Similarly in Fig. 212, the oats from B, which is 65° from the vertical, are almost clear, the mass at D having been discharged from bucket C.

Sizes of Head Pulleys for Centrifugal Discharge Elevators.—From what has gone before it is evident that there are three factors which determine the size of head pulleys. (1) That relation between diameter and revolutions per minute which gives a clean discharge according to Tables 35, 36 and 37 and which gives a belt speed sufficient for the quantity of material to be handled. (2) the necessity of having the diameter of the pulley at least five inches for each ply of the belt as discussed in Chapter V and Chapter XX. (3) In cases where the size of the foot wheel is limited, a ratio between diameters of head pulley and foot pulley which will permit the buckets to fill properly, see pages 218 and 219.

CHAPTER XVI

ELEVATOR BUCKETS

Discharge as Related to Shape of Buckets.—In Fig. 202, representing the conditions of discharge in high-speed elevators, the direction of the resultant for position 7 shows that if the bucket were made with the front parallel with the back there would be interference between the front and some of the grain leaving the bucket, and that the discharge would be delayed or some grain would be trapped within the bucket. In Fig. 203, representing the discharge at abnormally high speed, the resultants at 5, 6, 7, 8 and 9 are all directed toward the lip of the bucket; in Fig. 205, showing the discharge at abnormally low speeds, the resultants all point toward the back of the bucket, and in Fig. 206, which represents the conditions established by Table 36, the resultant is parallel with the back of the bucket and there is no tendency to trap the material.

These considerations show that at the higher speeds a clean discharge is favored by having the top angle T of a bucket relatively small (Fig. 215) and the bottom angle B relatively large, so that the bucket presents a wide-open mouth for the release of its contents. But on the other hand, the bucket lip must not be too low, nor the bottom angle too large, or some of the material will be spilled from the bucket at position 3 (Fig. 202). The shapes of various styles of centrifugal discharge elevator buckets on the market represent compromises between these opposing factors. Of all the buckets for elevating grain, the Buffalo bucket (Fig. 216) is most generally used for high speeds; it has a top angle of 80° and a bottom angle of 20°. Empire buckets have a top angle 83° and a bottom angle 15°; they are large in cross-section, but will not discharge at the speeds of Table 35. Rialto buckets with a top angle of 70° and a bottom angle of 30° or 40°

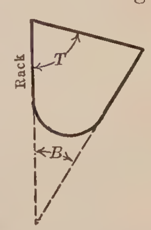


Fig. 215.—Typical Round-bottom Bucket for Centrifugal Discharge.

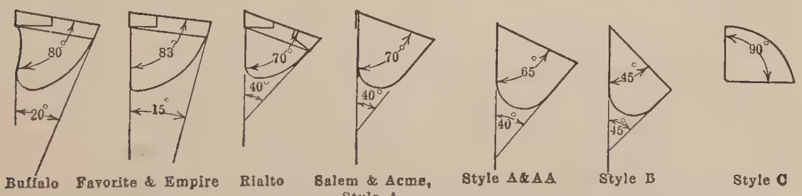


Fig. 216.—Angles of Various Sheet-steel and Malleable Iron Elevator Buckets.

will discharge at speeds even higher than those of Table 35, but for the same over-all dimensions they hold less than Buffalo buckets. The Minneapolis (Fig. 217) bucket, which is often sometimes run at very high speeds, is in cross-section approximately an equilateral triangle with straight sides and 60° angles. It will give a clean discharge at speeds higher than standard, but it is more apt to spill when the rising bucket meets the head wheel because the resultant pressure at that point has the direction *AB* at standard speeds. For still higher speeds the resultant *AC* is greater in amount and even more unfavorable in direction (see "Effect of High Speeds," p. 213) and acts to squeeze still more grain over the front lip. It is doubtful whether there is any gain in capacity by increasing the speed of these buckets more than 20 per cent over the figures given in Table .35 At still higher speeds the spill increases rapidly, the discharge is scattering, and much of the power consumed by the elevator is spent in forcing the buckets through the boot at excessive speed and in lifting more grain than is delivered to the chute at the head. In one elevator equipped with these buckets the head pulley is 72 inches, makes 56 r.p.m. with a belt speed of 1055 feet per minute; here centrifugal force is 3.2 times gravity. At the 24-inch foot pulley it is 9.6 times gravity. If a diagram for these conditions is drawn similar to Fig. 204 it will be seen that the resultant pressures are practically radial, the buckets fill entirely on the vertical run and begin to discharge as soon as they meet the head pulley.

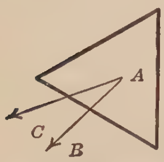


Fig. 217.—Resultant Pressures in "Minneapolis" Bucket on Meeting the Head Pulley.

Malleable-iron buckets in Style *A*, Fig. 216, have a top angle 65° and a bottom angle 40° and are well suited to handle nearly all rough and heavy materials at the speeds given in Table 36. For materials that are wet and stringy, or which stick to the bucket, a bucket, like Style *B*, Fig. 216, with a lower front will give a cleaner discharge. For material like raw sugar, which is very sticky, the wide-open mouth of the Style *C* bucket (Fig. 216) gives a better discharge and the material is not so apt to pack tight in the bottom.

The discharge at the head of a centrifugal discharge elevator is affected also by the shape of the bottom of the bucket; since this point is related to the spacing of the buckets it is discussed below.

Discharge as Affected by Spacing of Buckets.—Fig. 213 shows the discharge from bucket *B* striking the bottom and front of bucket *C*, which is 1 foot ahead of it, but if the buckets were 2 feet apart there would be no such interference, because in Fig. 213 the direction of the discharge from *B* shows that the mass would clear bucket *D* 2 feet ahead of *B* even at the low speed of 180 feet per minute. At higher speeds the buckets may be closer together, and as may be seen from Fig. 212 the spacing for the conditions shown might be even less than 1 foot and still the discharge from bucket *B* would clear the bucket ahead of it.

A comparison of Figs. 202 and 203 shows that as the speed increases, the material is thrown more nearly in a radial direction from the wheel and

with less chance of interfering with the leading bucket. The shape of the bucket also has a bearing on the spacing; if the bottom is sharp or of small radius and lies close to the belt, the discharge from the following bucket is not so likely to strike it, and if the discharge should strike it, a straight bottom is less likely to scatter the discharge than a full, rounded bottom. This explains why buckets of the Minneapolis type (Fig. 221) can be placed close together, almost touching, on a belt and run at high speeds; the mouth is wide open to permit a discharge similar to that shown in Fig. 203, and if the discharge from one bucket does hit the bottom of the one ahead, the grain is not scattered, perhaps hardly deflected from its path.

Buckets of the Buffalo type (Fig. 216) have a bottom sharper and less rounded than Empire buckets and hence can be placed closer together without making a scattering discharge. Fig. 218 shows to scale Buffalo buckets 8 inches deep, 8 inches projection, spaced 13 inches on a grain elevator belt passing over a 96-inch head pulley that makes 27 revolutions per minute. This speed is according to Table 35 and the conditions of discharge are shown in Fig. 202. The resultants drawn from positions 6 and 7 represent the direction of the arrows shown in Fig. 218, and the buckets 13 inches ahead of those positions are shown dotted. It is probable that some of the discharge from 6 hits the sloping bottom of the bucket 13 inches ahead and glances off, but the discharge is most active at 7, where the resultant pressure is greater, and so directed that the grain will clear the bucket 13 inches ahead of position 7. The 13-inch spacing gives satisfactory results with the 8 by 8-inch Buffalo buckets in the particular elevator which Fig. 218 illustrates, and the amount of spill is very small.

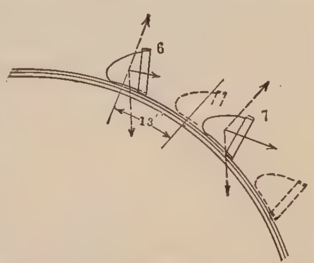


FIG. 218.—Discharge of Grain from 15 ×8-inch Buffalo Buckets Spaced 13 Inches Apart.

Kinds of Elevator Buckets. Seamed Buckets.—In elevators for grain, flour-mill products, etc., where the abrasion is not severe, it is customary to use light sheet-steel buckets; most of them are of the three-piece style with reinforcing band. The body sheet of tin plate or light sheet steel No. 24 or No. 26 gauge, is bent to form and fastened by lock seams to the two end pieces. The reinforcing band around the top edge, about 1 or $1\frac{1}{2}$ by $\frac{1}{8}$ inches in section, forms a digging edge in front and a hold for the bucket bolts in the back and gives the bucket the necessary stiffness to resist distortion without excessive weight. In high elevators run at high speed it is important to avoid unnecessary weight in the buckets; a heavy bucket costs more, adds to the load on the belt, and under the influence of the centrifugal forces existing in high-speed elevators it is more likely to be torn loose from the belt or pull the bolts through the belt.

Seamed buckets are sold by a number of manufacturers under trade-names which are well known in this country.

The Buffalo bucket (Fig. 219) is made in sizes from 12 to 20 inches long, with a brace for added stiffness in the sizes over 15 inches. The back is usually curved so as to match, to some extent, the bend of the belt as it wraps around the pulley. This has some advantage in reducing the pull on the bucket bolts in picking up the load (see Fig. 243), but at the same time it concentrates the pressure between the back of the bucket and the belt on two spots, and belts are sometimes cut through at those places. When the buckets are made with a flat back, the wear on the belt is more distributed.

Fig. 219.—Buffalo Buckets.

Rialto buckets (Fig. 220) are made in sizes up to 20 inches long; they will discharge clean at high speed, and because of their shallow form and relatively large bottom angle (see p. 225) they can be placed closer together on a belt than Buffalo buckets.

Buckets of the styles known as Empire, Favorite, etc. (Fig. 216), are made in a great variety of sizes from 5 to 20 inches long. They have bottoms more fully rounded than other styles and will nominally hold more for the same over-all dimensions. Because of their depth and the shape of their bottoms they must be spaced relatively further apart to avoid interference in discharge, and hence the net carrying capacity is not greater, even under conditions favorable for fully loading the buckets. They are not often used in high-speed elevators (Table 35), but rather in elevators of moderate capacity where the speed is within the limits of Table 36. They are, therefore, well suited to handle flour, bran, chaff and other fine, dry, light materials, as well as grain.

Fig. 220.—Rialto Bucket.

" Minneapolis " buckets are made in all sizes up to 20 inches. Fig. 221 shows the usual construction, sheet-steel body of No. 18 or No. 20 gage with a binding strip around the top.

Pressed and Riveted Buckets—One of the oldest buckets on the market is the Salem bucket (Fig. 222). It is a one-piece bucket with a front nearly straight, rounded bottom, and corners riveted together or spot-welded on the back. In the lighter gauges a reinforcing strip is folded over the top edge of the back. The front or lip is not usually reinforced. In heavier gauges the back strip is not used, in which case the central portion of the back is, as it were, depressed below the ends of the back by the thickness

FIG. 221.—Minneapolis Bucket.

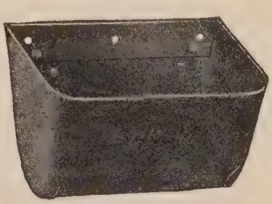


FIG. 222.—Salem Bucket.

of the metal, since the ends are bent around on the outside of the back and spot-welded or riveted there. This compensates to some degree for the crown of the pulley (see p. 261) when these buckets are fastened to a belt.

Salem buckets are made in sizes from 2 by 2 inches to 24 by 8 inches in various thicknesses of metal, No. 16 and heavier, and as to shape, either the standard high front, Style *A* (Figs. 222 and 223) or special low front, Style *B* like Fig. 224. These low-front buckets discharge more readily than Style *A*, but they hold much less on vertical lifts. There is not so much difference in inclined elevators, because the *B* bucket can carry a high surcharge

FIG. 223.—Style A Acme Bucket.

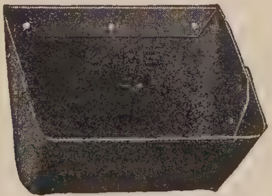


FIG. 224.—Style B Acme Bucket.

without spilling it over the front lip. For that reason, and because they discharge readily at low speeds, they are often used in inclined elevators when the angle of inclination from the vertical is 30° or more.

Acme buckets (Figs. 223 and 224) are like Salem buckets in shape, but the riveted or welded lap is on the ends instead of on the back. They are made in sizes 2½ by 2½ inches to 20 by 8 inches, Styles *A* and *B*.

In the lighter gauges, Salem and Acme bucket are used in moderate-speed elevators for grain and light mill products, pulverized chemicals and similar materials; in the heavier gauges they are used for materials heavier than grain, but they will not handle coarse, heavy substances so well as malleable-iron buckets.

Seamless sheet-steel buckets, known also as Caldwell or Avery buckets, are made of one piece of soft sheet-steel pressed in dies similar in form to Salem buckets. The corners are rounded, and the bucket is stiff enough for work in grain and similar materials. It is, however, more expensive than most three-piece buckets because of the heavier metal used. As compared with light buckets with a reinforced top edge, seamless buckets dig their load more easily and can be run with less power.

Comparative Capacities.—Table 38 shows that the carrying capacities of various styles of grain buckets do not differ much when expressed as cubic feet per foot of belt, because the buckets of fuller cross-section must be spaced further apart to get a clean discharge.

TABLE 38.—CARRYING CAPACITY OF GRAIN ELEVATOR BUCKETS PER FOOT OF BELT

Style	Length×Projection×Depth Inches	Contents of One Bucket, Cubic Feet	Spacing, Inches	Cubic Feet per Foot of Belt
Salem	20×7×7	.33	14	.28
Buffalo	20×7×7	.33	13	.30
Rialto	20×7×6½	.28	12	.28
Empire	20×7×7	.36	16	.27
Minneapolis	20×7×7¾	.23	8½	.32

FIG. 225.—Manufacturers' Standard Malleable Iron Buckets.

Malleable-iron Buckets.—Prior to 1908 there was no uniformity or regularity of sizes and shapes of malleable-iron buckets as made by different manufacturers. In that year the present Manufacturers' Standard Sizes

were established; the older styles and sizes have since become obsolete. The standard *A*, *AA*, *B* and *C* buckets shown in outline in Fig. 216 are illustrated also in Fig. 225. The bottoms are rounded to a rather large radius and each end is flared outward at a slope of 6°; since there are no seams or rivets, the buckets fill and discharge readily and material is not apt to stick in them. The metal is thicker than in most sheet-steel buckets, the corners are filleted and thickened; hence malleable-iron buckets are stiffer and stronger than most steel buckets of corresponding size and shape and they suffer less from distortion and abrasion in service. They are also less affected by rust.

Style A.—Over 75 per cent of the malleable-iron buckets sold in this country are Manufacturers' Standard Style *A*; they are used for coal, ores, chemicals, ashes and similar coarse materials. Table 39 gives weights and dimensions of the regular sizes.

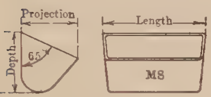


TABLE 39.—MANUFACTURERS' STANDARD MALLEABLE IRON BUCKETS—STYLE A

Principal Dimensions, Inches				Maximum Contents		Weight of Bucket, Pounds (Average)
Length	Projection	Depth	Thickness on the Flat (see note)	Cubic Inches	Cubic Feet	
4	$2\frac{3}{4}$	3	.078	16	.009	.90
5	$3\frac{1}{2}$	$3\frac{3}{4}$	.078	36	.021	1.23
6	4	$4\frac{1}{4}$	.093	55	.032	2.15
7	$4\frac{1}{2}$	5	.093	85	.049	2.68
8	5	$5\frac{1}{2}$	.093	115	.066	3.22
10	6	$6\frac{1}{4}$	.109	204	.118	5.40
11	6	$6\frac{1}{4}$	.109	223	.129	5.61
12	6	$6\frac{1}{4}$	.109	246	.142	6.53
12	7	$7\frac{1}{4}$	.140	332	.192	8.78
14	7	$7\frac{1}{4}$	.140	391	.225	10.90
14	8	$8\frac{1}{2}$	.172	509	.294	13.81
15	7	$7\frac{1}{4}$	.140	425	.246	12.30
16	7	$7\frac{1}{4}$	.140	467	.270	13.30
16	8	$8\frac{1}{2}$	.172	593	.343	17.54
18	8	$8\frac{1}{2}$	.172	668	.387	17.80
18	10	$10\frac{1}{2}$	.203	1053	.609	28.20
23	7	$7\frac{1}{4}$	.140	732	.424	19.00
24	8	$8\frac{1}{2}$	.172	887	.513	23.00

NOTE.—Corners are 50 per cent thicker.

Style AA buckets are like *A* buckets with the addition of metal to thicken the digging edge and the front corners. (See Table 40.)

Style B buckets hold less than Style *A* buckets when compared on the basis of their contained volume in cubic inches; but their carrying capacity in ordinary elevators is even less because, at the speeds of centrifugal discharge (Table 36) they are apt to spill a portion of their contents over the

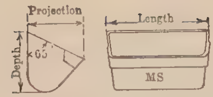


TABLE 40.—MANUFACTURERS' STANDARD MALLEABLE IRON BUCKETS—STYLE AA

Principal Dimensions, Inches				Maximum Contents		Weight of Bucket, Pounds (Average)
Length	Projection	Depth	Thickness on the Flat (see notes)	Cubic Inches	Cubic Feet	
6	4	$4\frac{1}{4}$	.093	55	.032	2.17
8	5	$5\frac{1}{2}$	.093	115	.066	3.55
10	6	$6\frac{1}{4}$	.109	204	.118	6.15
11	6	$6\frac{1}{4}$	.109	223	.129	6.66
12	6	$6\frac{1}{4}$	.109	246	.142	6.95
12	7	$7\frac{1}{4}$	.140	332	.192	9.70
14	7	$7\frac{1}{4}$	.140	391	.225	10.55
14	8	$8\frac{1}{2}$	.172	509	.294	16.30
15	7	$7\frac{1}{4}$	.140	425	.246	12.40
16	7	$7\frac{1}{4}$	.140	467	.270	13.08
18	8	$8\frac{1}{2}$	.172	668	.387	20.24
20	8	$8\frac{1}{2}$	.172	720	.417	26.50
24	8	$8\frac{1}{2}$	.172	928	.537	26.00

NOTE 1.—Corners are 50 per cent thicker.
NOTE 2.—Metal at digging edge is double thickness.

low front lip (see p. 213) if the elevator is vertical or nearly so when the bucket meets the head wheel. In inclined elevators, when the angle is more than 30° from the vertical, there is some risk that fully loaded *A* buckets will spill over the top edge. This is not so likely to happen with *B* buckets, hence these are sometimes used in such inclined elevators. They are of greatest use in handling clay and stringy or sticky materials in inclined elevators. These materials often stick in *A* buckets, but discharge more readily from the wide-open mouth of *B* buckets at the comparatively low speeds used in inclined elevators.

Less than 10 per cent of the malleable-iron buckets sold in this country are Style *B*; *A* buckets are more generally useful and will carry more material per dollar of investment.

Table 41 gives information about Style *B* buckets.

Style C buckets (see Table 42) are seldom used in belt elevators; they are suitable for wet sugar, damp clay and similar materials too sticky to discharge from *A* or *B* buckets.

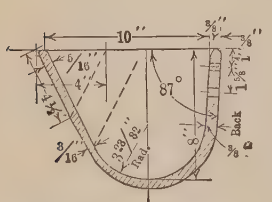


FIG. 226.—Cross-section of $17\frac{1}{2}$ by 10-inch Malleable Iron Bucket for Mineral Pulps.

Malleable-iron Buckets for Liquids and Pulps.—In order to prevent loss by splashing out over the front lip at the pick-up or on meeting the head pulley (see p. 220) it is an advantage in high-speed elevators to use buckets of special form with high fronts. Fig. 226 shows the cross-section of a $17\frac{1}{2}$ by 10-inch bucket used

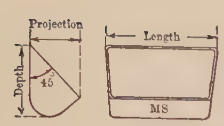


TABLE 41.—MANUFACTURERS' STANDARD MALLEABLE IRON BUCKETS—STYLE B

Principal Dimensions, Inches				Maximum Contents		Weight of Bucket, Pounds (Average)
Length	Projection	Depth	Thickness on the Flat (see note)	Cubic Inches	Cubic Feet	
4	$1\frac{1}{2}$	$2\frac{1}{4}$	.062	6	.0035	.41
7	$3\frac{1}{2}$	5	.078	55	.032	1.90
8	$3\frac{1}{2}$	5	.078	65	.038	2.01
10	4	$5\frac{1}{2}$	.093	107	.062	3.75
12	$5\frac{1}{2}$	$7\frac{1}{2}$	.109	233	.135	6.95
16	$6\frac{1}{2}$	9	.140	412	.238	13.15

NOTE.—Corners are 50 per cent thicker.

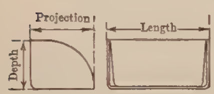


TABLE 42.—MANUFACTURERS' STANDARD MALLEABLE IRON BUCKETS—STYLE C

Principal Dimensions, Inches				Maximum Contents		Weight of Bucket Pounds (Average)
Length	Projection	Depth	Thickness on the Flat (see note)	Cubic Inches	Cubic Feet	
6	$4\frac{1}{2}$	4	.093	84	.049	1.93
7	$4\frac{1}{2}$	4	.093	98	.057	2.54
8	$4\frac{1}{2}$	4	.093	112	.065	2.50
10	5	4	.109	150	.087	3.75
12	5	4	.109	180	.104	3.90
14	7	$5\frac{1}{2}$	.140	437	.253	9.25
16	7	$5\frac{1}{2}$	.140	500	.289	10.80
18	8	8	.172	898	.52	17.65

NOTE.—Corners are 50 per cent thicker.

at a copper ore concentrating works in New Mexico. Other sizes are also made (see Table 45, page 238); these all have very thick backs for strength and to resist the abrasion from fine material which gets between the bucket and the belt. The fronts are also thickened.

Spacing of Commercial Buckets.—The parabolas 6 and 7 in Fig. 202 represent the discharge of free-flowing materials at the speeds given in Table 35, and buckets spaced as shown in Fig. 218 might be expected to give a clean discharge so long as the ratio of spacing to projection shown there was maintained. In practice, however, it is not advisable to space smaller buckets relatively so close as the 8 by 8-inch buckets shown in Fig. 218; the discharge is not quite so prompt, and so nearly radial, for several reasons. The smaller masses of material discharged from the buckets

meet more air-resistance, the friction between the material and the bucket walls is relatively greater and the sheet-steel buckets generally used in the smaller sizes have bottoms somewhat fuller and more rounded than the Buffalo buckets. For these reasons, buckets smaller than 8-inch projection must be spaced relatively further apart than Fig. 218 shows.

Table 43 gives closest spacings of sheet-steel buckets for the elevator speeds given in Table 35. Rialto buckets can be spaced close because they are shallower than other styles in proportion to their projection and hold less material.

TABLE 43.—FOR CENTRIFUGAL DISCHARGE ELEVATORS—CLOSEST SPACING OF BUCKETS FOR FREE-FLOWING MATERIALS AT SPEEDS GIVEN IN TABLE 35.

Style	Projection of Bucket												
	2	2½	3	3½	4	4½	5	5½	6	6½	7	7½	8
Acme or Salem	5	6	7	8	9	10	11	12	12	13	14	15	16
Buffalo	...	...	...	...	...	...	...	...	12	12	13	13	13
Rialto	...	...	...	...	...	...	9	10	10	11	12	12	12
Favorite, Empire Common Sense, Seamless	...	...	...	...	10	11	12	13	13	14	15	16	18
Minneapolis	½ to 1 inch clearance between buckets on the belt												

For the lower speeds of Table 36 buckets must be spaced farther apart. The reason may be seen in Figs. 202 and 203, as compared with Fig. 206. At the lower speeds the discharge is more nearly tangent to the sweep of the buckets, and more clearance is needed to avoid interference. Table 44 gives closest spacings of buckets for the elevator speeds given in Table 36.

TABLE 44.—FOR CENTRIFUGAL DISCHARGE ELEVATOR—CLOSEST SPACING OF BUCKETS FOR COARSE OR GRANULAR MATERIAL AT SPEEDS GIVEN IN TABLE 36.

Style	Projection of Bucket													
	2	2½	3	3½	4	4½	5	5½	6	6½	7	8	9	10
Acme, Salem or any of same form. Malleable iron, A, AA or B	6	7	8	9	10	11	12	13	14	15	17	20	22	24
Malleable Iron C	...	...	...	...	...	13	14	...	...	...	18	22	...	28

Pick-up as Related to Shape and Spacing of Buckets.—In handling lively free-flowing material like grain the buckets can be relatively close together, because the material flows into the buckets quickly; but for coal, minerals and coarse materials, the buckets must be further apart to allow time for filling. The figures given in Tables 43 and 44 are applicable to most materials; but if the buckets of a centrifugal discharge elevator are required to handle wet, stringy or sluggish substances, it may be necessary to space them even farther apart than Table 44 shows. An alternative is to use an elevator of a different type, one which picks up the material at a slower speed, because vertical centrifugal discharge elevators cannot be operated successfully at speeds much lower than those of Table 44. Inclined elevators, perfect discharge elevators on two strands of chain and continuous bucket elevators will discharge at very slow speeds and for some materials they are much more efficient than centrifugal discharge elevators.

Buckets with very low fronts, like Style *B* malleable-iron buckets, will not pick up a full load at speeds higher than Table 44. They are more useful in slow-speed elevators of other types, especially inclined elevators.

Style *C* malleable-iron buckets are not often used in centrifugal discharge elevators. When they are, they must be spaced far apart (see Table 44) so that the bottom does not interfere with the discharge from the following bucket. They will not pick up at high speed.

Capacities of Belt Elevators.—The capacities of elevators are usually stated as so many tons per hour, because that is the measure of the volume of material going through the plant per hour or per day, or the rate of unloading, storing, crushing, etc., in the operations with which the elevators are connected. So far as the elevating capacity of the buckets is concerned, the material lifted in one hour is sixty times the quantity lifted in one minute, but that is not always the case in the operations which the elevator serves. If the supply to a crusher, mill or machine is irregular, the feed to the elevator which takes away from the machine may for some seconds or minutes be at a rate much higher than the hourly rate which represents the working capacity of the plant. If there are under the machines hoppers or spouts large enough to store the excess quantity, they can be fitted with feeders or control gates to regulate the feed to the elevator, but most elevators are not equipped with feeding devices. Irregularity of supply to the elevator boot may come also from delays in car supply, interruptions in plant processes and chokes in chutes, which, when relieved, put an increased burden on the elevator for a time. See also page 148.

Peak-load Capacities.—For the reasons stated above, elevators should be designed for the *peak-load* or *per-minute* capacity rather than the average or per hour capacity. Neglect of this precaution leads to trouble and disappointment. An elevator seriously overloaded for less than a minute may fail for several reasons.

1. If the foot wheel is suddenly buried in material, the buckets are unable to dig their way out, the belt slips and the elevator stops.

2. If the supply to the boot is beyond the elevating capacity of the buckets the feed chute may fill up and choke.

3. When the elevator belt slips, its speed falls off to the point at which the buckets do not make a clean discharge, the spill falls down the back leg and adds to the accumulation in the boot, and the elevator stops.

There are many belt elevators in service which are large enough to give the required number of tons per hour with regular feed and steady operation, but too small to handle the load as it is delivered to the boot *per minute.* Such elevators have trouble with chokes, belts worn, torn or dried out by repeated slipping, pulley lagging worn off, buckets tearing off, and the annoying delays and expense of shut-downs and repairs.

Bucket Capacity and Elevator Capacity.—Under favorable conditions of pick-up and discharge, the capacity of an elevator in pounds per minute equals

$$\frac{\text{Bucket capacity in pounds} \times \text{Belt speed in feet per minute}}{\text{Bucket spacing in feet}}$$

but where the bucket does not pick up a full load or discharge clean, the capacity is less. For reasons given in Chapter XV, buckets in centrifugal discharge elevators take a full load only when the belt speed and diameter of foot pulley are correctly related and when the boot is of the right shape. In many elevators these favorable conditions do not exist, the pulleys in the boot are too small, the loading is too low and the buckets go up partly empty.

Manufacturers' catalogues rate buckets by their contents in cubic inches; this is the contained volume measured to the line AA (Fig. 227), but for reasons stated above, buckets in centrifugal discharge elevators do not usually fill to that line. In high-speed grain elevators when the conditions of loading and discharge are favorable it is proper to deduct 10 to 15 per cent in calculating the lifting capacity of the bucket. If the feed is such that the buckets can not complete their loading at or above the level of the foot shaft, the deduction should be greater than 15 per cent.

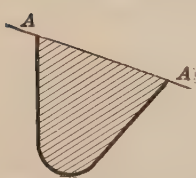


Fig. 227.—Theoretical Filling of Elevator Bucket.

Grain elevator buckets run at the speeds of Table 36 over good-sized foot wheels are likely to fill within 10 per cent of nominal capacity.

In ordinary centrifugal discharge elevators for coal, ores and minerals the buckets should not be expected to carry more than 75 per cent of their rated capacity. If the feed is too low or the foot wheel too small, the capacity will not reach 75 per cent.

When inclined elevators are run at the speeds of Tables 36 or 37 the pick-up is good and the buckets may load to nearly full capacity if the inclination is between 20° and 30° from the vertical.

When buckets dig from an open pile, filling will not average over 50 per cent of nominal capacity.

In handling liquids or mineral pulps at the speeds of Tables 35 or 36 it is hardly safe to count on the buckets in vertical elevators loading more than one-third full. Inclined elevators at the speeds of Table 36 will carry more, especially with a large foot wheel.

Examples.—Buffalo buckets, 15 by 8 by 8 inches, rated, at 581 cubic inches, used in a high-speed elevator, never carried more than 14 pounds of wheat in service. This is equivalent to 537 cubic inches, a loss of 14 per cent in capacity.

In tests of a well-designed elevator for pulverized ore, with 14 by 8-inch malleable iron, Style *A* buckets, the contents averaged 400 cubic inches per bucket, a reduction of 21 per cent from the nominal capacity of 509 cubic inches.

Table 45 gives facts about belt elevators in use in a copper smelter in New Mexico, most of them in wet service. The nominal capacities in the last line of the table are calculated on the assumption that every bucket takes a full load for every minute of the twenty-four hours. The great

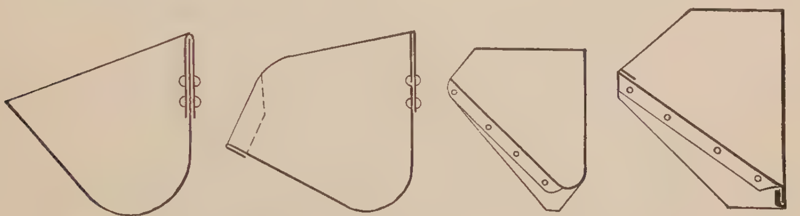


FIG. 228. FIG. 229. FIG. 230. FIG. 231.

EUROPEAN BUCKETS FOR GRAIN ELEVATORS.

FIG. 228.—"Deep-pressed" Bucket.
FIG. 229.—"Shallow-pressed" Bucket.
FIG. 230.—One-piece V Bucket with Rounded Bottom.
FIG. 231.—Two-piece V Bucket with Sharp Bottom.

margin between these figures and the actual capacities in ore, over and above the water handled, represents allowances for various items: (1) some irregularity in ore supply; (2) loss of time in shut-downs; (3) imperfect filling of buckets; (4) some spill on the lift and at the discharge into the head chute; (5) using throughout the mill, belts and buckets of standard sizes (that is, standard for that mill). This brings about a desirable uniformity among the elevators, although some are much too large for the work.

European Buckets for Flour and Grain.—The ordinary "deep-pressed" bucket used in Europe is a one-piece sheet-steel bucket similar to the Salem or Acme, and has a top angle of about 70° and a bottom angle of about 33° (Fig. 228). The "shallow-pressed" bucket is similar in construction to the "deep-pressed" bucket, but it does not match any American bucket in shape. It has a bottom angle of about 60° and the top angle measured from the back edge to the front lip is also about 60°; but the tops of the sides are carried up higher than the straight line which measures the top angle (Fig. 229). This bucket has several merits:

TABLE 45.—SPECIMENS OF ELEVATORS IN A CONCENTRATING WORKS IN NEW MEXICO

	Coarse Crushing Department	Coarse Crushing Department	Fine Crushing Departments				Flotation Department		
Elevator Number	1	2	3	4	5	6	7	8	9
Height of elevator	49′ 6″	59′ 6″	59′ 6″	60′ 0″	66′ 0″	66′ 0″	56′ 0″	53′ 0″	59′ 6″
Tons handled, 24 hours	6000	6000	2400	3200	200	2200	1250	250	150
Size of head-shaft, diameter × length	7″ × 8′ 5½″	7″ × 8′ 5½″	6½″ × 8′ 9½″	7″ × 7′ 11½″	4 3/16″ × 5′ 7⅜″	7″ × 8′ 5½″	7″ × 7′ 11½″	4 3/16″ × 5′ 7⅜″	4 3/16″ × 5′ 7⅜′
Size of head-pulley, diameter × face, inches	60″ × 38″	60″ × 38″	60″ × 28″	60″ × 32″	40″ × 20″	60″ × 38″	60″ × 32″	60″ × 28″	60″ × 26″
R.P.M. of head-pulley	29	32	30	28	38	30	30	28	34
Belt, width and ply	36″ × 11	36″ × 12	26″ × 12	30″ × 12	16″ × 10	36″ × 12	30″ × 12	26″ × 12	24″ × 12
Belt speed, feet per minute	449	510	471	431	404	460	468	438	534
Size of foot-shaft, diameter × length	3 15/16″ × 5′ 6½″	3 15/16″ × 5′ 6½″	3 15/16″ × 4′ 6½″	3 15/16″ × 5′ 0½″	3 15/16″ × 4′ 6½″	3 15/16″ × 5′ 6½″	3 15/16″ × 5′ 0½″	3 15/16″ × 4′ 6½″	3 15/16″ × 4′ 6½″
Size of foot-pulley	48″ × 38″	48″ × 38″	48″ × 28″	48″ × 32″	28″ × 20″	48″ × 38″	48″ × 32″	48″ × 28″	48″ × 26″
Batter or slope of belt on loaded side	24″	24″	24″	24″	24″	24″	24″	24″	24″
Buckets, length × projection	17½″ × 10″	17½″ × 10″	12″ × 8″	15″ × 9″	15″ × 9″	17½″ × 10″	15″ × 9″	12″ × 8″	12″ × 8″
Buckets, dead weight each, pounds	36	36	27	34	34	36	34	27	27
Buckets, cubic contents, each, cubic feet	.537	.537	.248	.429	.429	.537	.429	.248	.248
Buckets, spacing on belt	18″	18″	12″	16½″	16½″	18″	16½″	12″	12″
Buckets, double row (D) or single row (S)	D	D	D	D	S	D	D	D	D
Largest size of material	3″	2″	1″	.09″	.02″	.065″	.023″	.016″	.008″
Ratio of water to solids as handled	Dry	Dry	1½ to 1	2.2 to 1	12 to 1	4 to 1	4 to 1	9 to 1	12 to 1
Size of motor	Shaft	50 H.P.	50 H.P.	50 H.P.	15 H.P.	50 H.P.	50 H.P.	25 H.P.	25 H.P.
Nominal tons, 24 hours, based on full bucket capacity	28,000	30,000	21,000	24,000	11,000	28,000	25,000	20,000	24,000

1. In handling flour, bran or light mill products it will fill better because it has a wide-open mouth and the air can get out of the bucket and the material get into it without stirring up so much dust.

2. On account of the high ends it will carry more material than if the ends were cut off straight, as is usual in American buckets.

3. On account of the straight-line front and the large bottom angle it will discharge light mill products readily, and for the same reason there is a clean discharge of flour which tends to pack into small angles and sharp corners of buckets of some other styles. The low-front sheet-steel buckets of American makers generally have more of the front cut away; the top angle is therefore small and the bucket will discharge readily; but the carrying capacity is unnecessarily reduced for such substances as those mentioned, and for which sheet steel or tin buckets are in general use.

Buckets of angular form, like the Minneapolis bucket, are quite popular in Europe. They are made in one-piece style (Fig. 230) with a small fillet in the lower corner, or in two pieces with the bottom sheet separate, flanged and riveted to the ends (Fig. 231), in which case the bottom corner is sharp.

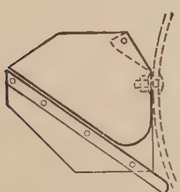


FIG. 232.

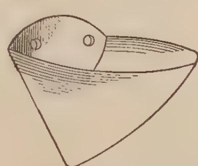


FIG. 233.

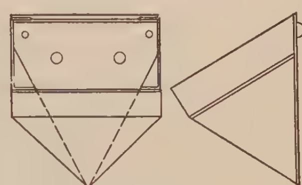


FIG. 234.

FIG. 232.—European Bucket for High-speed Grain Elevator.
FIG. 233.—"Funnel" Buckets for High-speed Grain Elevator.
FIG. 234.—High-speed Grain Elevator Bucket, set Continuous on the Belt.

Fig. 232 shows still another form of this bucket (U. S. patent 665273 of 1901) in which the lower edge of the front sheet is flanged and bears against the belt as a prop for the bucket. In all of these, the lower edge of the front must come close to the back, so that the discharge from the following bucket will either clear, or be deflected by, the front sheet.

European makers list sheet-steel buckets with a flat back and a kind of half-conical front (Fig. 233). These will dig through a deep mass of grain in a boot more easily than buckets of any other shape; they are light and strong and discharge well at speeds even higher than those of Table 35. They can be set with little clearance on the belt and will not interfere with the discharge. A bucket of this style 10 inches wide, $7\frac{1}{4}$-inch projection holds .16 cubic foot; at 10-inch spacing this is equivalent to .19 cubic foot per foot of belt. By comparing this with Table 38, page 230, and halving the figures of the latter for a bucket 10 inches wide, it is evident that these so-called "funnel" buckets will give a large capacity.

A similar bucket with a pyramidal bottom, and made from one piece of sheet steel, is shown in Fig. 234. It is covered by U. S. patent 788590 of 1905.

CHAPTER XVII

CONTINUOUS BUCKET ELEVATORS

Continuous Bucket Elevators.—When buckets of the shape shown in Fig. 201 are set close together on a belt they empty by a pouring action and do not need the throw imparted by high centrifugal force. Hence they can be run at comparatively slow speeds with merely enough velocity to dislodge the material from the bucket, to avoid dribble into the gap between the buckets and to assist the discharge to flow promptly over the bottom of the leading bucket.

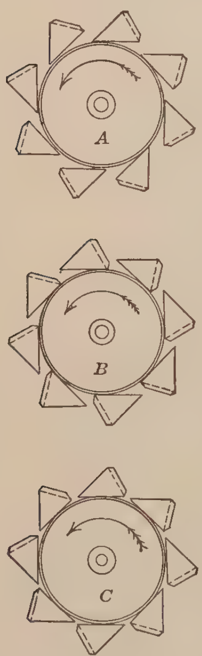


FIG. 235. — Pick-up and Discharge of Continuous Buckets as Affected by Method of Fastening to Belt.

Pick-up and Discharge.—The bolts which fasten the back of a bucket to a belt must necessarily be in one or, at most, two rows so as to allow the belt to bend on the pulleys (see p. 256). In fastening the comparatively shallow and round-bottomed buckets used in centrifugal discharge elevators, the bolts are near the top of the back edge, but with the deeper and heavier buckets used in continuous bucket elevators, the bolts must be about half-way between top and bottom in the back plate. In Fig. 235 the top half of diagram *A* represents continuous buckets bolted near the top edge, passing over the top of a pulley, while the bottom half shows the action under a foot pulley. The pick-up under the pulley might be called good, but the discharge is bad, because material will be poured out of one bucket into the space between the belt and the bucket ahead. In diagram *B*, showing buckets bolted at the bottom, the pick-up is bad, because, material will enter the gap between belt and bucket, but the discharge is good. If the buckets are fastened at the middle, according to standard practice (see diagram *C*), the result is a compromise; the gap which opens between the belt and bucket is relatively small and at the discharge point material is not likely to get into it unless the speed is so slow that the contents of the bucket dribble out. Material would get into the gap if the bucket picked up its load from a boot; for that reason, belt elevators of this kind should never be loaded from a boot, but from a chute (Fig. 236) at a point above the foot pulley where the buckets

are on the straight run, lie close to the belt and do not present any gap into which material might enter. When material does get between the belt and the back of the bucket it may wedge tight there and put a severe strain on the bolt fastening, and is apt to wear or punch holes in the belt from repeated pressure in going over the pulleys.

The distance x (Fig. 236) from the lower edge of the loading chute to the shaft in its upper position of take-up travel should be at least equal to the height of one bucket; more is preferable, to make sure that the bucket A will catch all that B misses, and that A is flat against the belt when the feed pours into it. At the head, the upper edge of the discharge chute is usually set at 45° below the level of the head shaft and as close to the elevator as the sway and movement of the buckets will permit; then if the buckets are properly shaped and run at proper speed, stone, coal, gravel and such materials handled in continuous bucket elevators will flow out in a clean discharge. If the material, gravel, for instance, is quite wet and contains sand, it is better to put the chute lower by a foot or two to catch the delayed discharge and the drip.

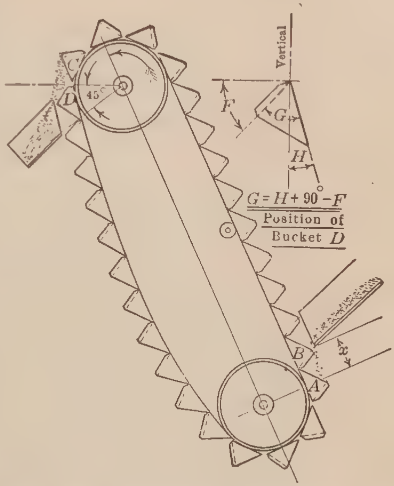


FIG. 236.—Loading and Discharging Continuous Bucket Elevator.

Shape of Continuous Buckets.—In Fig. 236 the contents of C are pouring out over the bottom of D, and it is evident that a clean and prompt delivery to the chute depends, among other things, on the angle at which the bottom of D stands at the moment it acts as a chute. In most continuous bucket elevators there is some "throw" which helps material across D to the chute, and some of the discharge may enter the chute without touching D at all; nevertheless, there is always some spill onto the bottom of D which must slide off.

In order to make the angle F (Fig. 236) steep enough to let material slide easily, the bucket must have the right shape; the angle G in the bottom must bear a proper relation to the material handled and to the angle H representing the deflection of the bucket from the vertical at the position D. If the elevator stands vertical H is zero and $G = 90° - F$; if, for instance, the material is clean and fairly dry, it will flow on a steel plate when $F = 40°$; then the bucket should have a bottom angle of 50°. It is generally better to have G larger than 50°, so that material will not be so likely to wedge in the bottom corner, and since gravel, damp coal, stone, dust and similar substances flow more readily when F is 45°, it is an advantage to incline the elevator so that H is at least 10°. For this condition

$G = H + 90° - F$ or $10° + 90° - 45° = 55°$. If the elevator has a greater inclination from the vertical, H might be 15°, then the buckets could have a bottom angle G of 60°. The angle H can be taken from Table 57; strictly considered, the angles B in that table refer to belts without applied tension; when take-up tension is applied, B becomes less and H greater, but for practical purposes, the table may be used and H may be considered equal to $90° - B$.

Based on these considerations, as well as those referring to the loading, most continuous bucket elevators on belt are inclined from 15° to 30° from the vertical and have buckets with the bottom angle G from 50° to 60°. Some buckets are made with G equal to 70°, but they do not give a clean discharge unless the elevator is inclined at least 25° from the vertical and handle dry, clean, free-flowing material. For that case, H is about 10° or 12° and $F = 30°$ or 32°.

From a consideration of the way the feed enters the bucket from the loading chute (Fig. 236) it is evident that the buckets must be open in front. They would hold more if made with a short front sheet parallel to the back sheet, but such a front would interfere with the loading, would splash and spill the material and would soon wear out.

Fig. 237 shows various shapes of continuous buckets, No. 1 being the ordinary two-piece bucket, one sheet forming the back and ends, another sheet, flanged at each end, forming the bottom. The corresponding three-piece bucket, No. 2, has two end pieces riveted to one sheet which makes the back and bottom. In order to prevent scatter and spill at the loading chute, buckets are sometimes made like No. 3 or No. 4, but they have given trouble at times, by stones catching between the end plates of adjacent buckets, and the projecting corners of the end plates are apt to be knocked out of shape in handling heavy material. These mishaps are not likely to happen with buckets like No. 1 and No. 2, because the end plates do not butt against each other and the corners have been trimmed off. Buckets like No. 5 have been made with the idea of improving the discharge, as where C discharges onto the back of D (Fig. 236). The overlapping end plates confine the discharge and prevent it from scattering, but in course of time the overhanging corners of the plates are likely to get out of shape and then spoil the overlap, or interfere with each other. Buckets, in end view like No. 2 or No. 4, are sometimes made tapering, narrower at the top than at the bottom so as to overlap and thus prevent the discharge from spreading out too far sideways. They have some advantage in that respect, but, on the other hand, loading is not so likely to be clean as when the end plates of the buckets are all in one plane, that is, not tapered.

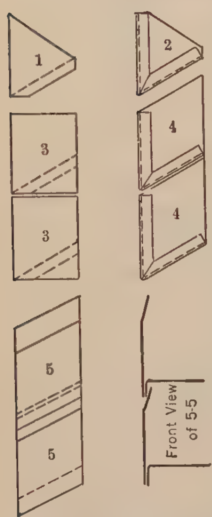


Fig. 237.—Some Forms of Continuous Buckets.

Height of Bucket and Diameter of Pulley.—In order that the bucket may not gap away from the belt too far (see diagram C, Fig. 235) the bucket must not be too high, measured along the belt, nor must the pulley be too small. Table 46 gives the amount of the gap in inches, measured radially, for various heights of buckets and diameters of pulleys. The gap should never be more than one-eighth the height of the bucket, and therefore combinations to the left of the heavy line in the table should not be used.

TABLE 46.—RELATION OF HEIGHT OF BUCKET TO DIAMETER OF PULLEY

Height of Bucket, Inches	Gap Between Belt and Bucket, Inches — Diameter of Pulley, Inches					
	18	24	30	36	42	48
8	.84	.64	.52	.43	.37	.33
10	1.29	1.00	.81	.68	.58	.51
12	1.81	1.41	1.15	.97	.84	.73
14	2.40	1.82	1.55	1.31	1.13	1.00
16	3.04	2.42	2.00	1.69	1.47	1.29
18	3.72	3.00	2.49	2.12	1.84	1.63

Buckets more than 16 inches high are seldom used, even with large pulleys; it is hard to fasten them securely to the belt with two or even three rows of bolts.

Capacities of Continuous Buckets.—It is not possible to give rules for the carrying capacities of continuous buckets. The loading chute is always narrower than the bucket and the ends do not fill so well as the middle of the bucket. The amount that can be piled in a bucket depends on the piling angle of the material, the shape of the front of the bucket and the inclination of the elevator; but it does not usually exceed 75 per cent of the maximum represented by the cubic contents of the bucket. It is generally necessary to make a sketch to determine the carrying capacity.

Speeds of Continuous Bucket Elevators.—On the use of continuous buckets for elevating grain at high speed, see page 226. Generally the term "continuous bucket elevator" is applied to machines that run at speeds lower than those of Table 36 and which do not depend altogether on centrifugal force to empty the buckets. The low limit of speed is that which will prevent material from dribbling into the gap between the buckets (see p. 244), and the high limit is determined by the nature of the material and the delivery to the buckets. Such elevators on belt are seldom, if ever, used to pick up material from a boot because of the danger that material will crowd between the belt and the bucket, pack there, and pull the bolts through the belt or tear the buckets off. When they are fed from a chute at some distance above the foot wheel, as they should be (see p. 241), the

considerations are that the bucket should have sufficient time to fill properly, and that the impact of the material delivered to the bucket should not be too severe for the fastening to the belt. If the material is large in proportion to the area of cross-section of the bucket, or if the pieces are long and flat, like shale or some kinds of crushed cement rock, the load does not settle quickly into position in the bucket and a relatively slow speed is necessary to load the bucket properly and avoid spill over the front edge. Elevators that handle such material may be run between 100 and 150 feet per minute.

Speed Must Not Be Too Low.—In handling materials, like moist fine ores or excavated earth or other substances which do not flow readily on themselves, continuous buckets may fill with a high surcharge. In passing over the head pulley the surcharge will spill into the gap between the buckets, unless the speed is high enough to influence the discharge by centrifugal action. For such conditions it is advisable to use belt speeds as high as those of Table 37, page 220.

Objections to High Speed.—If the material is small, like crushed slag or stone for road building, railroad ballast or concrete construction, the speed may be higher without much risk of spill and scant loading. Two hundred feet per minute has been considered standard for such work, but there are many belt elevators running at higher speeds which are thought to be satisfactory. In some cases the belt travels as fast as 300 feet per minute; the object, of course, is to get a high capacity from a belt and bucket of given size. Some of these elevators handle small stuff, relatively light in weight, like crushed slag, satisfactorily; but there are others carrying heavy, coarse materials, where the high speed causes an excessive amount of spill and unusual wear and tear on the belt and buckets. Buckets 15 inches high traveling 300 feet per minute pass a loading chute at the rate of four per second. Considering the depth of material in the loading chute, the time of loading such a bucket is perhaps as much as one-third of a second, certainly no more. This may be sufficient for fine material to run into and fill a bucket, but not so with coarse stuff. There are many elevators handling stone 2 inches and larger where the buckets do not take a full load and where the capacity elevated would be greater if the belt speed were less. Lower speed means better filling, less spill and scatter into the pit at the foot of the elevator and less pull on the bucket bolts at the loading point and in going under the foot wheel and over the head wheel.

Sizes of Pulleys.—For reasons given in Chapter V and Chapter XX the diameter of head pulleys should be at least 5 inches per ply of belt. Pulleys smaller than this do not grip the belt so well, and the belt wears out sooner because of slip or by reason of excessive stress on the friction rubber or the stitching which holds the plies together.

Loading Legs for Continuous Bucket Elevators.—In order to reduce the amount of spill and scatter, continuous buckets are sometimes run between side boards or plates at the loading point, or the two sides may be joined

by a front sheet below the loading chute so as to form a three-sided box a few feet deep. In some cases these have worked well, but in others they have been tried and then thrown out. In stone elevators, the material is apt to get into the clearance spaces between the moving buckets and the fixed plates, and either wear out the plates or damage the buckets and the belt. The fastening of a continuous bucket to a belt is necessarily confined to a number of bolts that perforate a narrow strip across the width of the belt; anything that puts an added strain on this section of the belt is to be avoided. If the loading chute is properly sloped and set with reference to the buckets, and if the belt speed is not too great, the amount of spill is ordinarily not objectionable. In most cases it is better to clear away the spill regularly rather than try to prevent it by the use of a loading leg.

For further information about inclined continuous bucket elevators see Chapter XXII. Refer also to Chapters XVIII and XX.

CHAPTER XVIII

BELTS FOR ELEVATORS

Belts for Elevators.—So far as materials of construction and methods of manufacture are concerned, elevator belts are like conveyor belts, and what is said in Chapter III applies generally to belts for elevating as well as for conveying.

Elevator service may be considered an extension of conveyor service; a belt will convey certain material on the level or on any slope up to 20°; fitted with cleats to prevent the material from rolling or sliding, it will carry up to, say, 30°; fitted with buckets to hold the material it will carry at any angle up to the vertical.

There are some features of elevator service which make that work harder for belts than conveyor service:

1. Most elevators are under 75-foot centers, very few are over 150-foot centers; they are shorter than conveyors, and since the speeds are not much different, the belt makes more contacts with the pulleys and is bent oftener.

2. Elevator pulleys, especially foot pulleys, are apt to be smaller, considering the number of belt plies, than corresponding pulleys on conveyors; the tendency to stretch or break the bond between the plies of fabric is therefore greater.

3. The unit stress in the belt, that is, the pounds pull per inch per ply is often greater in an elevator belt than in a conveyor belt (see p. 209).

4. Elevator belts are subject to cutting and wear on the outer side from material delivered against it by feed chutes or in the boot; they are cut by the top edge of the back of the bucket and worn by bits of material caught between the belt and the bucket. On the pulley side they are often gouged and torn by the heads of the bucket bolts or by material falling between the belt and the foot pulley. Belt creep and belt slip wear the belt (see p. 276) by rubbing on the head pulley; the same thing happens when dirt and grit adhere to the pulley side of the belt; water enters through the bolt holes and destroys the cotton fiber and the bond between the layers of fabric.

5. A conveyor does its work in the open; the manner of loading can be seen and the condition of the belt observed. An elevator is usually enclosed and the pick-up is not visible. It often happens that loose buckets and loose bolts are not noticed in time to prevent damage to the belt.

6. An overloaded conveyor belt gives warning by spilling the excess over the sides where it can be seen. A choke in an elevator boot is usually hidden from view; it may cause the belt to slow down or stop and before

the drive can be stopped, the head pulley, continuing to turn, may damage the belt or wear it through.

7. Elevator belts are more easily overloaded than conveyor belts. The normal ratings of conveyor belts are about one-half the maximum loading (see Fig. 137). Elevator capacities are often calculated from the cubic capacity of the buckets as given in manufacturers' catalogues. These may be realized if the pick-up is good, but where, as often happens, the foot wheel is too small or the boot is not suited to the material handled, the buckets do not take a full load. It is a matter of observation that an elevator with a steady feed to the boot will run with a series of buckets only partly full; then as the accumulation in the front of the boot piles up to the center of the foot wheel or higher, the buckets fill full for some seconds; then the loading falls off again and the cycle is repeated. The result is that the capacity of the elevator is less than what was calculated, and the margin between the actual and the calculated capacity may be so little that a slight excess of feed is apt to choke the elevator, with resulting injury to the belt.

The first belt elevators were used for grain in flour mills; in recent years their use has been extended to practically all kinds of elevating work except dredging.

Grain Elevator Belts.—Oliver Evans, in his "Miller's Guide," published in Philadelphia in 1795, describes as one of his inventions, what was then a new thing, at least to the milling business, i.e., an elevator composed of a leather belt with cups or buckets fastened on at intervals and discharging their contents by *centrifugal action* over the upper pulley. To handle 300 bushels per hour, the elevator consisted of a strap of harness leather 4½ inches wide with buckets holding 1 quart strapped on to the belt every 15 inches. The head pulley was 24 inches in diameter and made about 30 revolutions per minute. The foot pulley was smaller and was contained in a wooden boot which was part of a wooden casing with double legs. The buckets were of willow wood ⅜ inch thick, steamed and bent to form the front and ends; a piece of leather tacked on formed the bottom, and the elevator belt acted as the back, the elevator being inclined 15° or 20° from the vertical. Evans showed also how to make buckets of sheet iron, but that material was scarcer than willow wood in the United States in 1785, when the first of these elevators was built.

Evans and his successors, Ellicott and others, built many belt elevators in flour mills up to 1830; in 1842 when Joseph Dart built the first bulk storehouse ("elevator") for grain on the Great Lakes at Buffalo, it had a belt elevator of 1000 bushels per hour capacity. By 1866 many "elevators" had been built at Cleveland, Toledo, Chicago, Milwaukee and other Lake ports, some of them holding over a million bushels. The first "elevator" on the Atlantic coast was erected in Philadelphia between 1859 and 1863; it had elevators with leather belts 20 inches wide and ½ inch thick. Rubber belts came in about 1870 and became popular during the rapid building of grain "elevators" in this country during the seventies and eighties. Some

of these belts were of excellent quality (see p. 30), but as to their construction, the specifications for the early elevators give little information; those for the Pennsylvania Railroad Company's Girard Point Elevator in Philadelphia in 1881 merely call for " best quality smooth-surface gum belts."

With the growth of the business came the detailed specification for rubber belts, such as the well-known Metcalf specification (see p. 31). This specification and others similar to it are still in use; but as has been stated (p. 37), there is a growing dependence upon the quality which experienced manufacturers have put into their trade-marked belts and less insistence upon some of the details of the older specifications. What the purchaser really wants in a grain elevator is a belt that will last many years without separation of the plies; it is not possible to get this by a specification that merely calls for so many pounds friction test, because a high friction test does not necessarily mean a rubber that will last a long time before it loses its elasticity and tenacity (see p. 36).

Rubber belts for grain elevators, as now made, are generally of 32-ounce duck, 5, 6 or 7 plies thick according to the service. For most cases it is safe to use a " friction surface " belt, that is, one which has only the thin layer of friction rubber on the outside surface (Fig. 40), but where the grain is handled wet, as in oats bleachers, a rubber cover $\frac{1}{32}$ or $\frac{1}{16}$ inch thick all over is necessary. In the past, rubber-covered belts were standard for all grain elevator legs, but since the work is dry and not abrasive, a rubber cover is not highly essential. Some experienced buyers prefer to spend their money for quality of friction rather than for rubber covers on leg belts. The fact is that rubber belts with a low-grade friction do not hold together unless they have the protection of a rubber cover. The friction in " competition " belts is apt to have a high percentage of mineral matter, and consequently a poor bond between the plies of duck; with these belts, the standard rubber cover, which is $\frac{1}{32}$ or $\frac{1}{40}$ inch thick, serves a useful purpose in keeping out atmospheric moisture; and it prolongs their life. In belts of better grade, the friction is compounded to maintain its elasticity for a long time; it forms a good bond between the plies of duck and does not need the protection of a cover in the ordinary work of elevating grain.

The trade-marked belts made for grain elevator work by experienced manufacturers are not usually sold on the maker's specifications as to weight of duck or quality of friction, but it is understood in the trade that they are equal to or better than Stewart's specifications (see p. 37).

On the use of frictions that test still higher, see page 38.

Rubber Belts for Other Service.—Many good conveyor belts are made from 28-ounce duck, but elevator belts are seldom made from duck lighter than 32 ounces. A standard 32-ounce duck may have in the warp 23 threads per inch, 7 yarns per thread, and in the filler 13 threads per inch, 6 yarns per thread. Belts for heavy service may have 34-ounce or 36-ounce duck; the heaviest belts are built from 42-ounce duck.

As has been stated (p. 53), the weight of duck is not in itself a measure of the strength or worth of the belt; those qualities depend also upon the proportion of warp and filler threads, the twist of the threads, and the manner in which the plies are held together by the friction compound. The skill and knowledge of the belt manufacturer in combining these with the proper grade of friction and with the right covers, when necessary, determine the value of a belt for a particular service and its ability to withstand, to an economical degree, all those strains, shocks, cuts, punctures and other distresses to which elevator belts are liable.

Friction-surface belts have on their outer surfaces, only that thin layer of friction rubber which is calendered or pressed into the duck before it is assembled and cured. They can be used economically for some fine dry materials like crushed ores, pulverized dry coal, dry chemicals, etc.; but where the material is damp, as coal often is from exposure to rain or snow, or where the ore is wet, as from jigging or as handled in the flotation process of ore separation, then a rubber-covered belt is preferable for several reasons:

1. The rubber cover keeps moisture from penetrating the cotton fabric.
2. It forms a cushion to prevent the fine particles which stick to the belt and pulleys from being pressed into the fabric.
3. When the belt slips and creeps on the head pulley as it always does (see p. 276) the rubber protects the fabric from being worn away by the grit always present between the belt and the pulley.

Wear on Elevator Belts is internal and external. If the material handled is clean, dry, not abrasive, not lumpy, belts may fail because the friction dries out and the plies come apart; if the material is clean and wet, the plies separate sooner because water enters through the bolt holes or through cuts and cracks; and the external wear is also a factor because the belt is more likely to slip on the wet pulleys. If the material is sharp and cutting as well as wet, the external wear is more rapid, and if the pieces are also hard and large, internal and external wear are both more serious from the cuts and punctures which the belt receives.

Adding Covers to Elevator Belts is a means of equalizing the external and internal wear; it produces a balanced construction which prolongs the life of the belts. In that respect it is like adding covers to conveyor belts. Rubber belts were not a success for conveying coarse materials heavier than grain until Robins made them with rubber covers. Similarly, in many elevators handling ore, and especially wet ore, the cost of upkeep has been greatly reduced by the use of belts with rubber covers on one or both sides. In other words, in spite of the greater cost of belts with rubber covers, the cost of elevating one ton of material has been reduced and less time has been lost in shut-downs for repairs and replacements of belts.

Rubber-covered Belts.—The ordinary $\frac{1}{32}$ or $\frac{1}{40}$ inch of rubber which characterizes the cheapest rubber-covered belt serves its purpose when it is required merely to keep out atmospheric moisture, when the material is not lumpy or abrasive and when the slip of the belt on the head pulley is

comparatively slight. Where conditions are bad in these respects, a thicker cover is needed to make a balanced construction. On the pulley side, a rubber cover maintains a good contact with the head pulley in spite of dirt and grit; if the work is dry, it increases the coefficient of belt contact; if the work is wet, the coefficient may not be any greater than between a wet pulley rim and a friction-surface belt, but the cover certainly acts as a protection to the fabric when the belt creeps and slips. A cover on the pulley side acts also as a cushion to prevent injury to the fabric from hard pieces jammed between the belt and the foot pulley.

It also forms a cushion into which the heads of the bucket bolts can sink without tearing the fabric, and the heads are not so likely to come into contact with the pulley rim. When the bolt heads project beyond the belt surface and bear against an iron pulley rim, the belt is apt to slip, especially if wet. If the head pulley is lagged, the lagging may be cut and torn by the bucket bolts. Conversely, the cover on the pulley side of the belt protects the fabric from being cut by the lagging bolts or rivets which often project when the lagging wears thin. This may occur from the natural creep of the belt even though the belt may apparently not slip (see Fig. 250 and p. 277).

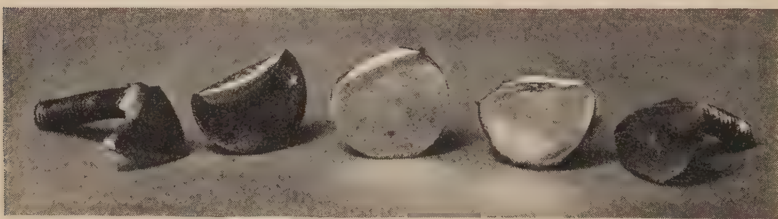


Fig. 238.—Lagging Bolts Worn by Slip and Creep of Elevator Belt.

Fig 238 shows a group of bolts used to fasten rubber lagging to the rim of a head pulley. The bolt heads have been worn away by the slip and creep of the elevator belt.

A rubber cover on the bucket side of the belt helps in several ways.

1. It protects the fabric from wear caused by direct impact of material against the belt in the boot or from a feed chute.

2. It resists the tendency of the forward edge of the bucket to cut the belt, either from hanging forward on the down run or from the action of centrifugal force in passing around the wheels.

3. It prevents the bits of material which catch back of the bucket from being forced into the fabric.

Figs. 41 and 42, page 23, show two 8-ply elevator belts, one with the ordinary $\frac{1}{32}$-inch rubber covers on each side; the other has $\frac{1}{32}$ inch on the pulley side, $\frac{1}{8}$ inch on the bucket side and a protection for the edge made by carrying the top cover around to the under side.

Thickness of Rubber Covers on elevator belts has been a matter of trial and investigation for several years. The size of the ore, its sharpness,

and whether it is wet or dry, the size of pulleys used, the size and spacing of buckets—all these are factors which determine the best proportion of belt plies and belt covers which makes a balanced construction for a particular elevator. These factors differ in various plants, and therefore the belt specification for the best service and lowest cost of handling will also differ.

Examples of Current Practice are given below:

1. An elevator 67-foot centers for cement clinker, size 1 inch and under, 60 pounds per cubic foot, temperature 160° F. Steel buckets 20 by 12 by 8 inches every 17 inches on the belt. Head pulley 48 inches, foot pulley 36 inches. Belt 22 inch, 8-ply, 36-ounce duck—the manufacturer's best grade, cover on pulley side $\frac{1}{32}$ inch, cover on bucket side $\frac{1}{16}$ inch. Speed 400 feet per minute = 32 r.p.m. of head pulley. Belt joint lapped and bolted. Belt lasted 450 days, working twenty-four hours a day. Life and service considered very satisfactory.

2. An elevator 57-foot centers, inclined about 10° for lead ore, size $\frac{3}{8}$ inch, wet, 100 pounds per cubic foot. Steel buckets 14 by 7 by $5\frac{1}{2}$ inches every 22 inches. Head pulley 42 inches, foot pulley 30 inches. Belt 16-inch, 8-ply, 36-ounce duck—the manufacturer's best grade, cover on pulley side $\frac{1}{32}$ inch, on other side $\frac{1}{16}$ inch. Speed 385 feet per minute = 35 r.p.m. of head wheel. Belt fastened with Jackson fastener. Belt lasted 682 days, elevated 455,000 tons; cost of belt per ton of ore .076 cent. This is a very good record.

3. An elevator 48 foot-centers, inclined about 15°, for rejections from jigging ore, size $\frac{3}{8}$ inch and under, damp, 160 pounds per cubic foot. Steel buckets 22 by 7 by 7 inches, spacing not stated. Head pulley 31 inches, foot pulley 24 inches. Belt 22-inch, 7-ply, 36-ounce duck—the manufacturer's best grade but not rubber covered (friction surface). Speed 520 feet per minute = 64 r.p.m. of head wheel. Belt lasted 126 days; cost per ton of material elevated .14 cent. This was better than the record of other belts used in this elevator, but the life was not so long as it might have been had the belt been rubber-covered and run at a slower speed. Five hundred and twenty feet per minute means 83 r.p.m. of the 24-inch foot wheel; entirely too fast for the pick-up of such material (see p. 212). The pulleys were too small in diameter for 7-ply belt (see Chapter V).

4. An elevator 55-foot centers, inclined slightly, for wet ore, $\frac{1}{2}$ inch size, 100 pounds per cubic foot. Steel buckets 14 by 7 by $5\frac{1}{2}$ inches spaced 18 inches, head and foot pulleys 36 inches. Belt 15-inch, 6-ply, grade not stated; covers both sides $\frac{1}{16}$ inch, belt joint, Jackson fastener, belt speed 470 feet per minute = 50 r.p.m. of head pulley. Belt lasted 632 days, twenty-four hours per day. A good record; belts seldom lasted over one year in this elevator. Speed is too high for a 36-inch head wheel.

5. An elevator 58-foot centers, inclined about 10°, for wet ore, size $\frac{3}{8}$ inch, 100 pounds per cubic foot, steel buckets 14 by 7 by $5\frac{1}{2}$ inches every 18 inches, head and foot pulleys 36 inches. Belt 15-inch, 7-ply, 36-ounce duck—the manufacturer's best grade, rubber-covered, thickness not stated, speed 470 feet per minute = 50 r.p.m. of head pulley. Belt lasted 416 days,

handled 436,000 tons; belt cost per ton elevated .08 cent. Considered a good record; the user expects belts to last ten or twelve months on this elevator. Speed is too high for a 36-inch head wheel.

6. An elevator 65-foot centers, inclined about 25°, for crushed stone, size 3 to 8 inches, dry, 100 pounds per cubic foot. Buckets 34 by 16 by 14 inches continuous on belt. Head pulley 48 inches, foot pulley 48 inches. Speed 250 feet per minute = 20 r.p.m. of head wheel.[1] Belt 38-inch 10-ply, 36-ounce duck—the manufacturer's best grade friction-surface belt, no rubber covers. Lasted four years eight months and carried between 1,500,000 and 2,000,000 tons of stone. An excellent record, the best ever made on that elevator.

Choice of Elevator Belts.—The kind of belt best suited to a particular elevator can be guessed at by some knowledge of what has given good service under similar conditions elsewhere; but since operating conditions are never exactly alike, the question can be settled in a purchaser's mind only by trial. The true test of an elevator belt is, "What does it cost per year or per ton of material elevated"? There are places where a belt is always discarded on account of external injuries, which cannot be avoided without serious changes in methods or equipment and at too great expense, and in which the wear cannot be resisted by any degree of high quality in the friction, or duck, or cover of the belt. In such cases, to buy expensive belts is throwing money away. On this subject, see Chapter III.

In normal elevators, however, the ordinary causes of belt failure are well known, and they can be opposed successfully by using the right kind of duck or the proper kind of friction or covers suited to the work. The main thing is to have the belt so strong that it will not fail suddenly under an accidental overload, or a choke from buckets ripping off, or a stick falling into the boot, etc. A sudden shut-down through the breakage of a belt may cost more in lost output of product, and in emergency repairs, than the price of a good belt. When the right belt has been chosen, it should be inspected regularly, and it will give plenty of notice before it is worn out; then a new belt can be ordered at the proper time in advance and when the day comes to put it in place, the replacement can be made in an orderly routine way without interruption of service and at the least cost.

On the relation between bucket width and belt width, see p. 261.

On the relation between belt tension and belt thickness, see p. 274.

Wet Elevating.—Handling the semi-liquid pulps in the wet concentration of ores is hard work for belts. Ordinary belts do not last long; the coefficient of belt contact with wet surfaces, rubber to rubber or rubber to iron, is only about half that of a dry belt on a lagged pulley; the belt is apt to slip unless it is pulled tight, and that slip in the presence of the fine grit which sticks to belt and pulleys wears out the driving face of the belt. The fine sand in the pulp works into the fabric through cuts and cracks, the plies separate and sand-blisters form. The water, getting into the cotton, mildews it, and the plies come apart. For these reasons a

[1] This is not a centrifugal discharge elevator.

friction-surface belt is out of place in handling mineral pulps, and even an ordinary rubber-covered belt with its $\frac{1}{32}$ or $\frac{1}{40}$ inch of cover does not usually return good service for the money spent on it, because the thin rubber is soon rubbed off by the slip and creep of the belt combined with the gritty sand.

Covers for Wet Elevator Belts.—Experiments made by mining companies for several years past have demonstrated the value of comparatively thick covers on the pulley side of belts handling wet ores and mineral pulps. One copper company uses belts with $\frac{5}{32}$-inch cover on the pulley side and $\frac{3}{32}$-inch on the bucket side; another company uses covers $\frac{3}{16}$-inch and $\frac{3}{32}$-inch on pulley side and bucket side, respectively; a lead-mining company uses $\frac{3}{16}$-inch rubber on the pulley side and only $\frac{1}{32}$-inch on the

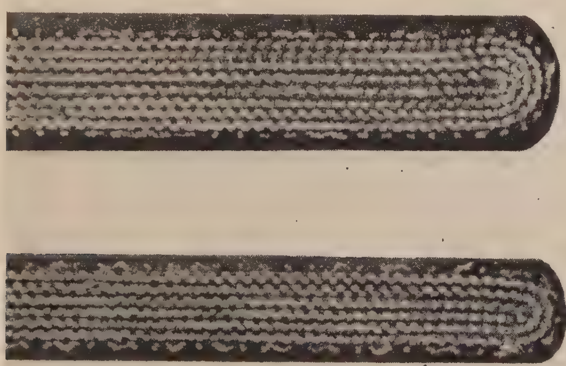


Fig. 239.—Elevator Belts with Rubber Covers on Both Sides. Covers Cemented and Vulcanized with Tie-gum Construction. (B. F. Goodrich Rubber Co.)

other side, depending on flaps of old belt (see Fig. 242), bolted under the buckets to protect the belt from cutting and abrasion.

Fig. 239 shows cross-sections of elevator belts made especially for handling wet pulps (B. F. Goodrich Rubber Co.). The 8-ply belt has $\frac{1}{8}$-inch covers on each side and the 6-ply belt has $\frac{1}{16}$-inch covers. The layers of friction rubber rolled into the duck are thicker than usual, so as to make the belt quite flexible in spite of its thickness and to keep water out of the fabric if the belt should be cut or the covers worn away. The covers are cemented to the body of the belt with the "tie-gum" construction referred to on page 25.

Width of Belts.—Belts for wet elevators should be wider than for dry elevating. There are several reasons for this:

1. The water, over and above what fills the voids in the crushed ore, must be provided for in choosing the size of the buckets, and hence the width of the belt.

2. Thin pulps are apt to splash out of the buckets in the pick-up and be forced out over the lip of the bucket by the resultant of the forces due to centrifugal force and gravity (see p. 220). Hence it is safe to use only a fraction of the nominal capacity of the buckets, as given in manufacturers' catalogues. Some experienced millmen use only one-third of the nominal capacity.

3. As between a narrow belt with buckets that project far from the belt, or a wide belt with buckets of less projection, the wide belt is preferable; it is less likely to slip on the wet pulley, and the pressures which tend to cut or wear the belt are less per square inch of belt surface.

See page 221 on the pick-up and discharge of mineral pulps.

Belts for Continuous Bucket Elevators.—The steel-plate buckets used in these elevators weigh more per foot of belt than buckets used in centrifugal discharge elevators; the backs are always flat, the projection generally

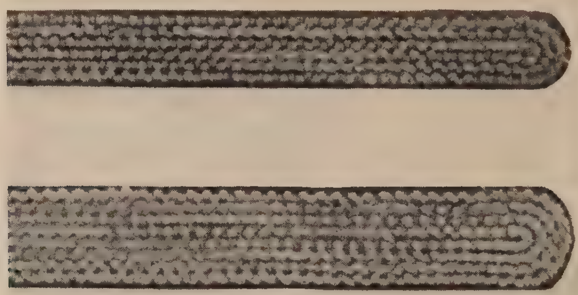


Fig. 240.—Elevator Belts with Heavy Duck for Hard Service. (B. F. Goodrich Rubber Co.)

greater than with round-bottomed buckets, and the distance A (Fig. 264) relatively smaller compared with the dimensions of the bucket. Consequently the pull on the bolts is severe, and unless the belt is stiff and strong it may be injured on the pulley side by the pressure of the heads of the bucket bolts; or the bolts may pull clear through the belt.

For these reasons, and because of the great wear on the belt surface, belts for heavy continuous bucket elevators should be made of duck heavier than the 32-ounce used in most rubber belts for elevator service. Rubber belts with 36-ounce and 42-ounce duck are made for heavy stone and ore elevators with continuous buckets, generally with thin rubber covers for dry work. Fig. 240 shows two specimens; the 6-ply belt has warp threads larger and fewer per inch than in ordinary elevator belts, and the filler threads are correspondingly closer and thicker. The 8-ply belt shows a very heavy close-woven duck.

Stitched canvas belts made of standard 32-ounce duck—37-ounce on the basis on which rubber belts are graded (see p. 46)—show great resist-

ance to the tendency of the bolts to pull through the belt, especially when saturated with a Class 1 drying compound (see p. 47) and properly stretched and cured. For strength of canvas belts see Chapter III.

Balata belts made of 38-ounce duck have a density and strength which fits them for work of this kind. For strength of balata belts see Chapter III.

CHAPTER XIX

FASTENING BUCKETS TO BELT

Fastening Buckets to Belt.—Since a belt, in passing around a pulley, forms an arc tangent to the flat back of a bucket, the bolts which fasten on the bucket should theoretically be in a single row across the back of the bucket. This is the rule on all elevators handling grain and similar materials: the bodies of grain buckets are made of tin plate or light sheet steel, No. 24 or No. 26 gauge, and the holes are punched in a single row through the fold of metal or reinforcing band at the top of the back. If the entire back were thick, as in malleable-iron buckets or in seamless steel buckets, it would be possible to get a better grip on the belt by using bolts in two rows an inch or two apart, but unless the pulleys are relatively large, or the rows close together, this practice will injure the belt or the bucket.

From Fig. 241, representing a belt on a pulley, it is evident that if the holes are punched in the bucket for two rows of bolts, and if the belt while straightened out is punched to match, the holes no longer match when the belt runs over a pulley; the holes in the belt are pulled out of parallel by the bend of the belt and a gap opens between the belt and the back of the bucket. The difference between the chord A and the arc B is not great (see Table 47), but for a 7- or 8-ply belt which is $\frac{1}{2}$ inch thick, the difference of the arcs B and C is more than $\frac{1}{16}$ inch for a row spacing of 2 inches on pulleys of 34-inch diameter or less. Bolt holes in buckets are made $\frac{1}{16}$ inch larger than the bolts and allow some freedom for the bolts; but too much movement of that kind is apt to cut off the bolts or wear out the backs of the buckets where the nuts bear against them.

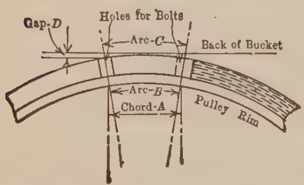


Fig. 241.—Double Row of Bucket Bolts Passing Over Pulley. (See Table 48.)

The gap D is $\frac{1}{32}$ inch or more for a row spacing of 2 inches in a belt as it bends over a pulley of 34-inch diameter or less. This dimension is a measure of the tendency of the bolts to bend the backs of the buckets or pull through the belt when passing over a pulley. There is a definite pull on these bolts when the bucket digs its load out of a boot, and when the material is hard and lumpy and the belt speed high it is not uncommon to find the belt on the pulley side injured by the movement of the bolts; or the bolts may even pull clear through the belt.

Continuous Buckets on Belt.—The large steel-plate buckets used on stone elevators are often so heavy that it is necessary to fasten them to

TABLE 47.—DIMENSIONS OF FIG. 241 FOR TWO ROWS OF BOLTS 2 INCHES APART

Diameter of Pulley. Inches	Arc B—Chord A Inches *	Arc C—Arc B For Belt ½ Inch Thick. Inches †	Gap D Inches *
12	.010	.167	.091
24	.0025	.083	.044
36	.001	.056	.030
48	.0004	.042	.022
96	.0002	.021	.010

* Proportional for spacings other than 2 inches (approximately).
† Proportional for thicknesses other than ½ inch.

the belt by two rows of bolts to prevent them from pulling loose or injuring the belt. These rows are midway in the height of the back of the bucket; whether they are 1 or 2 inches apart, the head and foot pulleys should not be less than 30 inches in diameter, and the loading should be arranged so that the buckets receive material direct from a chute while on a straight run and never have to dig out of a boot or even come into contact with material spilled under the foot wheel.

Malleable-iron Buckets on Belts.—In belt elevators handling ores and other gritty and heavy materials it is customary to use malleable-iron buckets; and when the projection of the lip from the belt exceeds 5 or 6 inches the pull on the bolts from the digging and from the weight of the bucket is often too much for a single row of bolts. In spite of the disadvantages of two rows of bolts, it is advisable to use two rows for buckets 10 by 6 inches and larger. If the rows are not over ¾ inch apart, the difference between B and C (Fig. 241) is less than $\frac{1}{32}$ inch for all belts not over ½ inch thick on pulleys at least 24 inches in diameter, and D is less than $\frac{1}{64}$ inch under the same conditions. For spacing of bolts see Table 49.

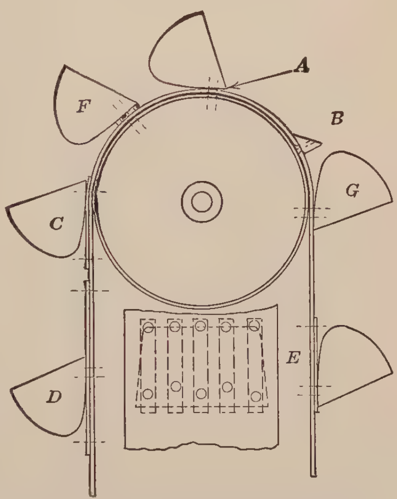


FIG. 242.—Devices to Protect Elevator Belts from Injury where Buckets are Bolted on.

At any rate it is better to hold the buckets on securely and avoid accident, even though the wear on the belt may be somewhat greater. It may be said, however, that the tendency to cut the belt at A (Fig. 242) may be less when the bucket is held by two rows of bolts. This is particularly true on the down run of belt; heavy buckets upside down tend

to swing away from the belt with the leading edge as a pivot (see G, Fig. 242). Some bits of material are always picked up in the boot where the bucket gaps away from the belt and carried between the belt and the back of the buckets ahead of the bolts. Any pressure there when the bucket is inverted may force particles into the belt and damage it, unless a pad of belting (see C, Fig. 242) is used back of the bucket or unless the bolts are kept tight.

Besides bolts, other methods have been used or proposed to fasten buckets to belts. Oliver Evans (see p. 247) used straps and buckles; Griscom, in 1896, patented a sheet-metal clip fastened to the bucket and bent over the edges of the belt. Another inventor proposed, in 1914, to fold the belt into a loop at each bucket and fasten the bucket to the loop. Filling and discharge of the bucket were apparently secondary items in this design. None of these schemes is in use at present; others could be mentioned, but they are not so cheap, simple and handy as the bolt fastening.

Pull on Bolts.—When a bucket with a flat back is fastened to a belt by a single row of bolts there is only a short contact between it and the belt as it bends around a pulley. When the strain of digging comes on a bucket there is a pull on the bolts which in Fig. 243 is measured by $T = \frac{RP}{A}$, where R is the resistance to the travel of the bucket through the boot. The value of T for a given bucket can be reduced by making the bucket with a curved back as in the Buffalo bucket (Fig. 219); then, in Fig. 243, $T = \frac{RP}{B}$. Since B may be two or three times A, the pull on the bolts is only one-half or one-third as much and where the digging is severe, as in grain elevator boots filled up above the level of the foot shaft, or at the foot of marine legs (Fig. 263) digging cargo grain, it is advisable to use buckets with the curved back.

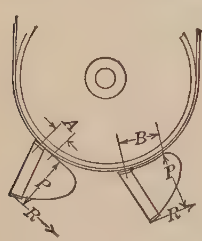


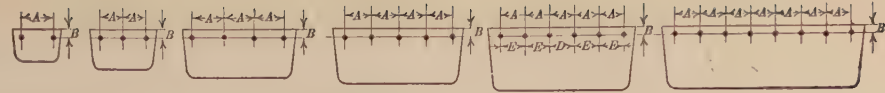
Fig. 243.—Pull on Bucket Bolts Dependent on Shape of Bucket.

To some extent, the same may be said of buckets fastened on by two rows of bolts. The bucket then has a longer contact with the belt and the pull on the bolts is less.

The elevator bucket shown in Fig. 232 is used on grain elevators in Europe; it has a flat back, but the sheet forming the bottom is extended and flanged at the lower end to act as a prop against the belt in going around the foot wheel. With this construction, the distance corresponding to A or B (Fig. 243) is even greater than in the Buffalo bucket, and the pull on the bolts is therefore less. At the same time, in going over the head wheel, the extended bottom sheet acts as a deflector for the discharge from the following bucket, and the buckets therefore can be placed close together or slightly overlapping.

Tables 48 and 49 (Weller Manufacturing Co.) give standard sizes and

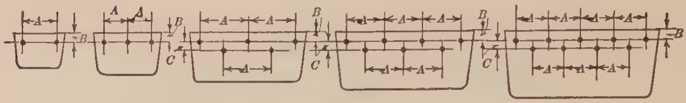
TABLE 48.—STANDARD ELEVATOR BUCKET PUNCHING. (For Attaching Buckets to Belts)



Size of Bucket, Inches	Favorite, Buffalo, Rialto						Minneapolis High-speed "V" Type						Salem			
					Bolt Holes						Bolt Holes		14-Gage and Lighter. All $\frac{9}{32}$ Holes		12-Gage and Heavier. All $\frac{11}{32}$ Holes	
	A	B	D	E	Number	Diameter	A	B	D	E	Number	Diameter	A	Number of Bolts	A	Number of Bolts
2×2	1	$\frac{5}{16}$	...	...	2	$\frac{3}{16}$										
$2\frac{1}{2} \times 2$	$1\frac{1}{2}$	$\frac{5}{16}$	...	...	2	$\frac{3}{16}$	...	...	...	...	...	...	$1\frac{1}{4}$	2		
$3 \times 2\frac{1}{2}$ and 3	$1\frac{1}{8}$	$\frac{3}{8}$	...	...	2	$\frac{9}{32}$	$1\frac{1}{8}$	$\frac{3}{4}$	...	...	2	$\frac{9}{32}$	$1\frac{3}{4}$	2		
$3\frac{1}{2} \times 2\frac{1}{2}$ and 3	$1\frac{5}{8}$	$\frac{3}{8}$	...	...	2	$\frac{9}{32}$	$1\frac{5}{8}$	$\frac{3}{4}$	...	...	2	$\frac{9}{32}$	$2\frac{1}{8}$	2		
4×3 and $3\frac{1}{2}$	$2\frac{1}{8}$	$\frac{3}{8}$	...	...	2	$\frac{9}{32}$	$2\frac{1}{8}$	$\frac{3}{4}$	...	...	2	$\frac{9}{32}$	$2\frac{1}{2}$	2		
$4\frac{1}{2} \times 3$ and $3\frac{1}{2}$	$2\frac{5}{8}$	$\frac{7}{16}$	...	...	2	$\frac{9}{32}$	$2\frac{5}{8}$	$\frac{3}{4}$	...	...	2	$\frac{9}{32}$	$3\frac{1}{4}$	2		
$5 \times 3\frac{1}{2}$ and 4	$3\frac{1}{8}$	$\frac{1}{2}$	...	...	2	$\frac{9}{32}$	$3\frac{1}{8}$	$\frac{7}{8}$	...	...	2	$\frac{9}{32}$	$3\frac{1}{2}$	2	3	2
$5\frac{1}{2} \times 4$	$3\frac{5}{8}$	$\frac{1}{2}$	...	...	2	$\frac{9}{32}$	$3\frac{5}{8}$	$\frac{7}{8}$	...	...	2	$\frac{9}{32}$	4	2	$3\frac{1}{2}$	2
6×4	$4\frac{1}{8}$	$\frac{1}{2}$	...	...	2	$\frac{9}{32}$	$4\frac{1}{8}$	$\frac{7}{8}$	...	...	2	$\frac{9}{32}$	$4\frac{1}{2}$	2	$4\frac{1}{4}$	2
$7 \times 4\frac{1}{2}$	5	$\frac{9}{16}$	...	...	2	$\frac{5}{16}$	5	$\frac{7}{8}$	...	...	2	$\frac{5}{16}$	$5\frac{1}{2}$	2	5	2
8×5	3	$\frac{5}{8}$	...	...	3	$\frac{5}{16}$	3	1	...	...	3	$\frac{5}{16}$	$3\frac{1}{4}$	3	3	3
9×5 and 6	$3\frac{1}{2}$	$\frac{5}{8}$	...	...	3	$\frac{5}{16}$	$3\frac{1}{2}$	1	...	...	3	$\frac{5}{16}$	$3\frac{3}{4}$	3	$3\frac{5}{8}$	3
$10 \times 5\frac{1}{2}$ and 6	4	$\frac{5}{8}$	...	...	3	$\frac{5}{16}$	4	$1\frac{1}{2}$	...	...	3	$\frac{5}{16}$	$4\frac{1}{4}$	3	$3\frac{7}{8}$	3
$10 \times 6\frac{1}{2}$ and 7	$2\frac{5}{8}$	$\frac{3}{4}$	...	...	4	$\frac{5}{16}$	$2\frac{5}{8}$	$1\frac{1}{2}$	...	...	4	$\frac{5}{16}$	$4\frac{1}{4}$	3	$3\frac{7}{8}$	3
11×6 and 7	3	$\frac{3}{4}$	...	...	4	$\frac{5}{16}$	3	$1\frac{1}{2}$	...	...	4	$\frac{5}{16}$	$3\frac{1}{8}$	4	3	4
$11 \times 7\frac{1}{2}$ and 8	3	$\frac{3}{4}$	...	...	4	$\frac{3}{8}$	3	2	...	...	4	$\frac{3}{8}$	$3\frac{1}{8}$	4	3	4
12×6 and 7	$3\frac{5}{16}$	$\frac{3}{4}$	...	...	4	$\frac{5}{16}$	$3\frac{5}{16}$	$1\frac{1}{2}$	...	...	4	$\frac{5}{16}$	$3\frac{1}{2}$	4	$3\frac{1}{4}$	4
$12 \times 7\frac{1}{2}$ and 8	$3\frac{5}{16}$	$\frac{3}{4}$	...	...	4	$\frac{3}{8}$	$3\frac{5}{16}$	2	...	...	4	$\frac{3}{8}$	$3\frac{1}{2}$	4	$3\frac{1}{4}$	4
13×6 and 7	$3\frac{5}{8}$	$\frac{3}{4}$	...	...	4	$\frac{5}{16}$	$3\frac{5}{8}$	$1\frac{1}{2}$	...	...	4	$\frac{5}{16}$	$3\frac{3}{4}$	4	$3\frac{1}{2}$	4
$13 \times 7\frac{1}{2}$ and 8	$3\frac{5}{8}$	$\frac{3}{4}$	...	...	4	$\frac{3}{8}$	$3\frac{5}{8}$	2	...	...	4	$\frac{3}{8}$	$3\frac{3}{4}$	4	$3\frac{1}{2}$	4
14×6 and 7	4	$\frac{3}{4}$	...	...	4	$\frac{5}{16}$	4	$1\frac{1}{2}$	...	...	4	$\frac{5}{16}$	$4\frac{1}{8}$	4	$3\frac{7}{8}$	4
$14 \times 7\frac{1}{2}$ and 8	3	$\frac{3}{4}$	...	...	5	$\frac{3}{8}$	...	2	3	$2\frac{1}{4}$	6	$\frac{3}{8}$	$4\frac{1}{8}$	4	$3\frac{7}{8}$	4
15×6 and 7	$3\frac{1}{4}$	$\frac{3}{4}$	...	...	5	$\frac{5}{16}$	...	$1\frac{1}{2}$	3	$2\frac{1}{2}$	6	$\frac{5}{16}$	$3\frac{1}{4}$	5	$3\frac{1}{4}$	5
$15 \times 7\frac{1}{2}$ and 8	$3\frac{1}{4}$	$\frac{3}{4}$	...	...	5	$\frac{3}{8}$	...	2	3	$2\frac{1}{2}$	6	$\frac{3}{8}$	$3\frac{1}{4}$	5	$3\frac{1}{4}$	5
16×6 and 7	...	$\frac{3}{4}$	3	$2\frac{3}{4}$	6	$\frac{5}{16}$	...	$1\frac{1}{2}$	3	$2\frac{3}{4}$	6	$\frac{5}{16}$	$3\frac{1}{2}$	5	$3\frac{3}{8}$	5
$16 \times 7\frac{1}{2}$ and 8	...	$\frac{3}{4}$	3	$2\frac{3}{4}$	6	$\frac{3}{8}$	...	2	3	$2\frac{3}{4}$	6	$\frac{3}{8}$	$3\frac{1}{2}$	5	$3\frac{3}{8}$	5
18×6 and 7	...	$\frac{3}{4}$	$3\frac{3}{4}$	3	6	$\frac{5}{16}$	...	$1\frac{1}{2}$	$3\frac{3}{4}$	3	6	$\frac{5}{16}$	4	5	$3\frac{1}{8}$	6
$18 \times 7\frac{1}{2}$ and 8	...	$\frac{3}{4}$	$3\frac{3}{4}$	3	6	$\frac{3}{8}$	...	2	$3\frac{3}{4}$	3	6	$\frac{3}{8}$	4	5	$3\frac{1}{8}$	6
19×6 and 7	$3\frac{3}{8}$	$\frac{3}{4}$	...	...	6	$\frac{5}{16}$	$3\frac{3}{8}$	$1\frac{1}{2}$	...	...	6	$\frac{5}{16}$	$4\frac{1}{4}$	5	$3\frac{1}{8}$	6
$19 \times 7\frac{1}{2}$ and 8	$3\frac{3}{8}$	$\frac{3}{4}$	...	...	6	$\frac{3}{8}$	$3\frac{3}{8}$	2	...	...	6	$\frac{3}{8}$	$4\frac{1}{4}$	5	$3\frac{1}{8}$	6
20×6 and 7	$3\frac{9}{16}$	$\frac{3}{4}$	...	...	6	$\frac{5}{16}$	$3\frac{9}{16}$	$1\frac{1}{2}$	...	...	6	$\frac{5}{16}$	$4\frac{1}{2}$	5	$3\frac{3}{8}$	6
$20 \times 7\frac{1}{2}$ and 8	$3\frac{9}{16}$	$\frac{3}{4}$	...	...	6	$\frac{3}{8}$	$3\frac{9}{16}$	2	...	...	6	$\frac{3}{8}$	$4\frac{1}{2}$	5	$3\frac{3}{8}$	6
22×6 and 7	4	$\frac{3}{4}$	...	...	6	$\frac{5}{16}$	4	$1\frac{1}{2}$	...	...	6	$\frac{5}{16}$	4	6	$3\frac{7}{8}$	6
$22 \times 7\frac{1}{2}$ and 8	4	$\frac{3}{4}$	...	...	6	$\frac{3}{8}$	4	2	...	...	6	$\frac{3}{8}$	4	6	$3\frac{7}{8}$	6
24×6 and 7	$4\frac{3}{8}$	$\frac{3}{4}$	...	...	6	$\frac{5}{16}$	$4\frac{3}{8}$	$1\frac{1}{2}$	...	...	6	$\frac{5}{16}$	$3\frac{1}{8}$	8	3	8
$24 \times 7\frac{1}{2}$ and 8	$4\frac{3}{8}$	$\frac{3}{4}$	...	...	6	$\frac{3}{8}$	$4\frac{3}{8}$	2	...	...	6	$\frac{3}{8}$	$3\frac{1}{8}$	8	3	8

$\frac{3}{16}''$ holes use $\frac{3}{16}''$ rivets; $\frac{9}{32}''$ and $\frac{5}{16}''$ holes use $\frac{1}{4}''$ bolts; $\frac{3}{8}''$ and $\frac{11}{32}''$ holes use $\frac{5}{16}''$ bolts.

TABLE 49.—MALLEABLE BUCKETS



Number of Holes	2 Holes		3 Holes				5 Holes			7 Holes		9 Holes	
Width of Bucket	4	5	6	7	8	9	10	11	12	14	16	18	20
A	3	3	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	$3\frac{1}{2}$	$3\frac{3}{4}$	$4\frac{7}{8}$	$3\frac{1}{4}$	4	$3\frac{1}{2}$	4
B	1	1	1	1	1	1	1	1	1	1	1	1	1
C	....	....	....	....	....	....	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$

Bolt holes in malleable buckets all $\frac{9}{32}$-inch diameter for $\frac{1}{4}$-inch bolts.

spacing of holes in sheet-steel and malleable-iron buckets for attaching to belts.

Damage to Belts where Buckets Are Bolted On.—Heavy steel or malleable-iron buckets fastened to belts and used to elevate hard, sharp ores, coke and minerals, often cut the belt at *A* (Fig. 242), especially if the bolts get loose and allow the bucket to move with relation to the belt. Bolts loosen from several causes:

1. Nuts slacking off.
2. Metal of the bucket wearing away under the nuts.
3. Belt wearing thin, or back of bucket wearing thin, from moving on each other.
4. Bolt heads wearing deep into the belt.

When the bolts get loose, centrifugal action at the pulleys tends to throw the bucket outward with the leading edge of the back as a fulcrum, and the pressure there drives hard particles into the belt; or the edge, if sharp, may cut the belt.

There is generally some wear between the belt and the flat back of the buckets used in elevating hard, gritty materials. Some of the discharge gets into the gap which opens there in going over the head wheel, or dribbles into it when the buckets are inverted on the descending run and hang away from the belt (*G*, Fig 242). The same thing happens at the foot wheel when the buckets gap away from the belt.

Protective Devices.—Various devices are in use to protect the belt from the wear mentioned above. A rubber cover on the belt is often the cure and may pay for itself in the longer life of the belt and the lower cost of repairs and renewals. In one elevator where the life of the belts averaged only a few months, triangular prisms of wood (*B*, Fig 242), equal in length to the width of the belt, were fastened on by wood screws below each bucket to prevent small bits from getting between the belt and the bucket. This scheme cost only a few dollars, but it more than doubled the life of the belt.

In the Western mining country where centrifugal discharge belt ele-

vators are extensively used for handling not only fine material, wet or dry, but also lump ore up to 3-inch size, it is common practice to use a pad of old belt between the bucket and the elevator belt (*C*, Fig. 242). Sometimes the bucket bolts do not go through the elevator belt, but only through the pad, which is then extended above and below each bucket to protect the belt from abrasion (*D*, Fig. 242). This is open to the objection that unless the bolts are kept tight, fine stuff will work in between the pad and the belt and, being confined there, will rub and injure the belt.

In wet elevators a narrow strip of old belt is used with each bucket bolt (*E*, Fig. 242) to space the bucket away from the belt and leave room for water to wash away the grit from behind the bucket. Soft rubber washers (*F*, Fig. 242) have been used for the same purpose and to keep the leading edge of the back from digging into the belt. Buckets mounted on thick washers cannot do heavy digging in a boot.

Troubles from Buckets Working Loose.—Besides the cutting of the belt mentioned above, there are other troubles, still more serious, due to nuts working loose and coming off the bolts. When only a few bolts in a bucket stay tight, the strain on them is excessive and they may damage the pulley side of the belt or pull through the belt, or break off. A bucket falling off into the boot may rip other buckets off and cause a breakdown; buckets thrown off into the head chute may damage machinery fed by the elevator, or cause spouts and hopper gates to choke. In grain elevators it is good practice to put a grating near the top of the head chute to catch such loose buckets, as well as the sticks of wood, pieces of paper, etc., which often pass from grain cars into the elevator leg. A clean-out door should be provided to give access to the screen for examination and cleaning (see Fig. 290).

Inspection of Buckets.—Trouble and expense can be avoided by inspecting buckets and their fastenings regularly and tightening or replacing the loose or broken bolts. If the belt shows signs of being cut or worn by the buckets or bolts new spots can be made to take the wear by shifting all the buckets to new positions somewhere between the old positions. Some men who install belt elevators punch the belt at the start for two or three settings of buckets; that is, if the buckets are to be spaced 18 inches apart, the new belt would be punched for bolts every 9 or 6 inches. This saves time and money when it becomes necessary to shift the buckets.

Width of Bucket and Width of Belt.—The usual crown of belt pulleys is $\frac{1}{8}$ inch on the diameter per foot of face—that is, the face of a pulley for a 12-inch belt is $\frac{1}{16}$ inch higher in the center than at the edges. When a belt runs over a crown-face pulley its center is stretched more than its edges, and a bucket fastened to the belt must either bend in the back or tend to pull the bolts through the belt; or else the belt does not conform exactly to the crown along the line of the bolts. In grain elevators the belts are relatively stiff and the buckets light and the back of the bucket springs slightly to match the crown when the face is not too wide. Sheet-steel buckets of heavier metal do not spring so easily, and malleable-iron buckets

are too stiff to spring at all. Hence it happens that when wide stiff buckets are fastened to a belt of the same width the bolts are apt to dig into and injure the pulley side of the belt. Instead of using malleable-iron or stiff steel buckets, say, 20 inches or wider on a belt of that width, it is better to use smaller buckets of half the width in double row to give the required capacity. The spacing between consecutive buckets in each row should, of course, be what will give a clean discharge at the head, and if the widths of the buckets in the two rows do not overlap along the center line of the belt there will be no interference with the discharge when the buckets are set staggered.

Buckets in Double Row.—In grain elevators and other elevators using wide belts it is now general practice to use double rows of staggered buckets when belts are wider than 22 inches (see Fig. 244). The advantages of this construction are as follows: (1) The buckets avoid the crown of the pulleys. (2) The buckets are stronger and stiffer and the fronts are less likely to pull out under heavy load, or on striking an obstacle like a stick of wood in the boot. (3) If a bucket is spoiled by any mishap, the cost of replacing it is less than if the accident happened to a bucket of double the width. (4) The work of pulling the buckets through a deep mass of material in the boot is less than for wide buckets in single row.

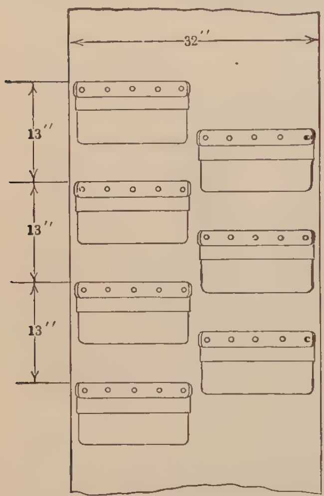


Fig. 244.—15-inch Buckets Spaced 13 Inches in Double Row on 32-inch Belt.

Belts in Casings.—When elevator belts run in casings it is usual to make the width of the belt 2 inches more than the width of the bucket, or 1 inch more if the bucket is not over 6 inches wide; then there is a margin of belt on each side which keeps the bucket from striking the casing. Of course casings are supposed to be made with clearance enough to avoid interference, but if they should twist or get out of line, it is better that the belt should rub than that the bucket should strike.

If there is no casing, or if the elevator is enclosed in a roomy housing, the margin of belt is not necessary as a guard; but where the head pulley gets wet or works in a cloud of dust, a belt no wider than the buckets is more likely to slip than a belt made a few inches wider. The coefficient of belt contact is less under such circumstances than when the head pulley is dry and free from dust.

Buckets to Match Crown of Pulleys.—A patent was granted in 1916 (No. 1194308) on a bucket with its back curved to match the crown of the pulley. Such buckets are not in practical use. The idea might be applied, with some advantage, to wide buckets, but, for reasons stated above, it

is better to use narrower buckets in two rows instead of very wide buckets in single row. Aside from that, there are mechanical difficulties in making sheet-steel buckets with the curve for crowning combined with the curve in the bottom of the bucket. It would be still more difficult to apply the idea to grain buckets with a curved back like Fig. 219, but malleable-iron buckets could be made with that feature if there were a demand for it.

Joining Ends of Elevator Belts.—The requirements for a good joint for elevator belts differ in some ways from those for conveyor belts. In the latter, the fastening must be flat on both sides because both sides of the belt run over the idlers. In elevator belts, the fastening need not be flush or even flat on the bucket side of the belt; the pull in the belt per inch per ply is often greater than in conveyor belts and a stronger splice is needed. Another difference is that the take-up in an elevator is usually shorter (see p. 292), resplicing must be done oftener, and a fastening that can be taken apart and put together readily in a confined space is preferable. For these various reasons, elevator belts are joined with bolted splices rather than with clinch hooks or clinch rivets.

One of the oldest and simplest fastenings is shown in Fig. 245. It is

Fig. 245.—Bolted Clamp Joint for Elevator Belt.

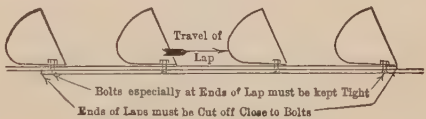


Fig. 246.—Bolted Lap Joint.

very strong and resists shocks, but it is not easy to apply to belts over 5-ply, and when it is put on thick belts, the bending in going over pulleys is localized at the corners of the flat bars, even though they are rounded. This leads to breaking the warp threads and to cracking the belt crosswise. This joint should not be used on canvas belts; they are too stiff and the duck is too heavy to stand the bending.

Lap Joints.—The joint used most frequently on grain elevator belts and for heavy continuous bucket elevators is a plain lap splice (Fig. 246). The lap may cover a distance of 4 feet or more and is held together by the bucket bolts. The joint is simple and strong and requires no extra parts; when the belt becomes too long, it is shortened by one or two bucket spacings, bolted together again, and the excess length is cut off. It is successfully used on 5- or 6-ply grain elevator belts, but on 8-ply belts, the double thickness at the lap measures about 1 inch, and when this bends over pulleys, the two thicknesses tend to move on each other and there are strains which work the bolts loose in the holes and cause wear on the belt under the bolt heads and nuts. If the laps are not held tight together, material gets between them, the bolts pull through the belt and the belt may break across the line of bolt holes. To prevent this, the bolts must be kept tight, especially those at the ends of the laps.

In stone elevators, sand and grit, getting between the laps, has been known to grind the belt partly through until it broke. The same thing has happened when an end of belt at the lap was not cut off close to the bolts; bits of stone wedged behind the projecting flap of belt and finally cut the belt so badly that it broke. In this particular case the belt was 12-ply and was lapped the wrong way for the travel of the belt. The right way is shown in the figure; the cut end of belt on the inside of the lap should trail over the pulleys, not run against them.

Butt-strap Joints.—The ends of heavy belts are frequently butted and the joint made by a piece of belting as wide as the elevator belt and two or three times as long as it is wide. In elevators with spaced buckets the length of the butt-strap may be twice its width and it is riveted on. In continuous bucket elevators it usually extends under two buckets on each side of the butt and is secured by the bucket bolts (Fig. 247). This joint has the same merits and the same defects as the lapped joint mentioned above; in addition, a bucket may have to be left off at the joint to make room for the double row of bolts where the ends of the belt come together. These bolts must be close to the cut ends of the belt to prevent material from working in between the belt and the butt-strap, and they must be kept tight.

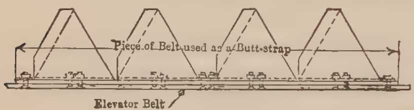


Fig. 247.—Butt-strap Joint.

The Jackson belt fastener (Fig. 248) consists of a series of stamped steel plates each with two countersunk head bolts, two oval cup washers with prongs, and two sleeve nuts. The ends of the belt are cut square with a thin piece of canvas belt as a templet; the bolt holes are punched in them. The bolts, with the oval washers on them, are inserted from the pulley side of the belt; then on the bucket side, the canvas templet is laid for a cushion and the plates are put on. When the nuts are screwed tight, the cup washers pull the belt up into the concaves of the top plates, the sleeve part of the nut wedges the warp threads together and the belt is held tight. The shape of the top plates is such that the two bolts do not stand parallel to each other, but at an angle to make them approximately radial in passing around pulleys of a size suited to the fastener. This prevents the bolts from moving in the belt or tearing it in passing over pulleys.


Fig. 248.—Elevator Belt Joined with Jackson Fasteners.

The Jackson joint is made for all thicknesses of belts; it is strong, and when applied to stiff thick belts, it is not apt to injure them as sometimes happens with lap joints or butt-strap joints. It is one of the best joints for stitched canvas or balata elevator belts.

Fig. 249 shows a butt-strap joint which has been used on belts handling very gritty material. The ends are joined by a metal fastener of some kind, and a strip of belt, usually lighter than the elevator belt, is placed over the joint and under the buckets to act as a cushion for the latter, to protect the metal fastener from the grit and also to strengthen the joint.

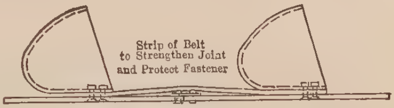


FIG. 249.—Elevator Belt with Bolted Fastener and Butt-strap.

CHAPTER XX

DRIVING BELT ELEVATORS

Drive by Head Pulleys.—In the design of a belt and bucket elevator the diameter of the head wheel must always be considered in connection with its speed; the two are not independent, but for a good discharge of materials, they hold rather definite relations to each other. These relations, for the usual working conditions, are given in Tables 35, 36 and 37, and the conditions for which it is proper to use these tables are described on pages 217, 218 and 219.

The theory of belt driving is based on the assumption that so long as the belt bends freely to the curvature of the pulley the driving effect is independent of the diameter of the pulley. This is confirmed in practice and by Haddock's experiments referred to in Chapter V. If the diameter of the pulley is at least four or five times the number of plies in the belt, the tractive effect does not vary with the size of the pulley; hence the general rule is that a 6-ply belt, for example, should have a head pulley at least 24 inches in diameter, better still 30 inches, and so far as the internal wear in the belt is concerned, the larger the diameter the better. It is, however, a fact that most elevator belts fail for reasons other than internal wear, and there are disadvantages in making the head pulleys too large; the head group takes up too much space, the head chute drops lower, the casing is larger, supports become heavier, larger driving machinery is needed, and for most of these items the cost is greater.

The driving contact between a head pulley and an elevator belt depends upon the angle of wrap and the coefficient of friction between the belt and the rim of the pulley. Usually the angle of wrap is 180°; hence the general expression for the ratio of tensions on the up-side T_1 and the down-side T_2 (see p. 109) becomes, for a coefficient of friction of .25, $\frac{T_1}{T_2}=2.19$ and for a coefficient of .35, $\frac{T_1}{T_2}=3.00$ (see Table 20).

That is, a head pulley will drive an elevator belt when the pull on the down side is from one-half to one-third the pull on the up side, assuming that pulley and belt are clean and dry. If the pull on the down side is more than one-half the pull on the up side, the drive is more certain to act under unfavorable conditions, such as dust, dirt on the belt, wet pulley face, pulley side of belt rough and torn, etc.

Coefficients of Friction.—The coefficients .25 and .35, mentioned above, are satisfactory in calculations for belt conveyors and generally give good

results in calculations for belt elevators also. The general formula for the value of $\frac{T_1}{T_2}$ assumes that f, the coefficient of friction, depends solely on the nature of the surfaces in contact. Experiments by Wilfred Lewis (Transactions A. S. M. E., Vol. 7) show that as the load increases and $T_1 - T_2$ becomes greater, the coefficient of friction also increases, although the belt creep and the belt slip increase at the same time. Table 50 abstracted from Carl Barth's comment on the Lewis experiments (Transactions A. S. M. E., 1909) gives in column 7 values of f for leather belts on clean iron pulleys at 800 feet per minute; they vary from .25 to .72.

These experiments were made with clean leather belts; but, reasoning from the known behavior of various kinds of belt in the transmission of power, there is no reason to think that rubber or other fabric belts would act differently, except for possible differences in the numerical values of the coefficient of friction. It is probable that the coefficients for these belts on clean iron pulleys or on pulleys covered or lagged with rubber are larger than .25 and .35, respectively, but when the belt is wet, as in elevators that handle wet ores, or when it becomes covered with granules of dirt, or when the interior of the casing is thick with dust as in grain elevators, then

TABLE 50.—VARIATION OF COEFFICIENT OF BELT FRICTION WITH SLIP AND CREEP ON DRIVING PULLEY

1 Experiment Number	2 Tension per Square Inch Belt at Rest. Pounds	3 Tension per Square Inch, on Tight Side. T_1 = Pounds	4 Tension per Square Inch, on Slack Side. T_2 = Pounds	5 Observed Loss in Slip and Creep Per Cent of Belt Travel	6 Calculated Loss Due to Creep Alone. Per Cent of Belt Travel	7 Coefficient of Belt Friction Corrected for Centrifugal Force
60	81.6	125.33	58.67	0.5	0.41	0.25
61	81.6	131.42	46.58	0.9	0.53	0.34
62	81.6	142.00	42.00	1.7	0.62	0.41
63	81.6	152.41	35.75	3.0	0.73	0.49
65	81.6	179.92	29.92	12.0	0.91	0.61
66	127.5	177.42	77.42	0.5	0.52	0.27
68	127.5	198.25	64.92	0.8	0.69	0.37
69	127.5	208.77	58.67	1.0	0.77	0.42
70	127.5	219.08	50.75	1.7	0.87	0.47
71	127.5	229.50	46.17	2.6	0.95	0.54
72	127.5	244.08	44.08	3.8	1.02	0.57
73	127.5	256.58	39.92	3.5	1.10	0.62
74	127.5	252.42	35.75	8.6	1.13	0.68
75	127.5	283.66	33.67	15.2	1.25	0.72
128	343.5	511.3	227.0	0.5	0.85	0.26
131	343.5	557.0	187.2	1.1	0.99	0.35
133	343.5	589.5	162.4	1.8	1.30	0.41
134	343.5	603.0	148.2	2.7	1.39	0.45
135	343.5	618.0	134.0	5.1	1.49	0.49

NOTE.—Since 1-ply thickness in a rubber or canvas belt is about $\frac{1}{16}$ inch, divide the figures in column 2 and 3 by 16, to estimate the equivalent tension per inch per ply in the belts, had they been of fabric instead of leather.

the coefficients are less than for clean pulleys. How much less we do not know. Haddock, in 1908, made experiments (Transactions A. S. M. E., Vol. 30) with a 12-inch 4-ply belt wrapped 180° on plain and rubber-covered pulleys dusted or coated with various substances; but his values (see Table 51) are quite erratic and can only be taken to show that the coefficients are less when the pulley rim is dusty or damp than when it is clean.

For dusty work the coefficients of friction for rubber or fabric belts on iron pulleys may be taken at .20, and for rubber-covered pulleys .27. For these values the corresponding ratios of $\frac{T_1}{T_2}$ are 1.87 and 2.33, respectively. For wet work, the coefficient may be called .20, whether the pulley is bare or lagged.

TABLE 51.—TRACTIVE EFFECT EXPRESSED AS PERCENTAGES BASED ON CLEAN DRY SURFACES OF RUBBER BELT AND IRON PULLEYS

(Haddock's Experiments, 1908—)

Condition of Contact Surfaces	Rubber Belt on Iron Pulley. Per Cent	Rubber Belt on Rubber Lagged Pulley Per Cent
Clean, dry	100	108
Clean, damp	92	101
Covered with dry coal-dust	92	80
Covered with damp coal-dust	76	52
Covered with dry clay	63	55
Covered with damp clay	55	65
Covered with dry slate-dust	68	138
Covered with damp slate-dust	85	147
Covered with dry, sharp sand	42	80
Covered with damp, sharp sand	62	80

It is certain that a pulley covered with rubber belt will pull more than a plain iron pulley if the work is dry, and although some engineers doubt the value of lagging on head pulleys of wet elevators, there are some practical advantages in it, even though it may not increase the value of f, the coefficient of belt contact. When belts wear down to the fabric on the pulley side and become rough, and when bolt heads project beyond the belt surface, then the contact with an iron face is not continuous, but interrupted, and the belt if wet is more likely to slip; but if the pulley is covered, the rim accommodates itself better to these irregularities and the driving contact is better. This is especially true if the lagging is not ordinary friction-surface rubber belt, but standard belt lagging which is several plies of fabric covered with a layer of rubber.

The application of the above to the design of an elevator can be discussed best by reference to an example.

Pull at Head of Grain Elevator.—The unbalanced torsional pull at the rim of the head pulley, which measures the power required to drive the elevator belt, is the sum of several items:

1. Weight of grain in the lifting buckets $= G$.
2. Drag of buckets through the grain in the boot $= B$.
3. Friction of foot shaft and pulleys $= F$.

The total pull in the elevator belt is the torsional pull plus the weight of belt and buckets on the rising side.

It is easy to calculate item G directly, but not B and F. F is small and may be estimated, but B must be derived indirectly from power tests of the elevator.

Calculation of Pull from Power Readings.—A certain grain elevator had a nominal capacity of 12,000 bushels per hour; the calculated capacity based on all buckets lifting their full load and discharging it without spill was 14,560 bushels per hour. When the following test was made, the elevator was lifting wheat at its regular rate; assuming a possible loss of 10 per cent in filling the buckets and in spill at the head, the work was probably at the rate of about 13,000 bushels per hour. The lift was 216 feet.

Power delivered to the motor			80 kilowatts
80 kilowatts $= \dfrac{80 \times 1000}{746} =$		107.2 h.p.	
Motor loss (efficiency about 94 per cent)	$= 6.4$ h.p.		
Estimated loss in silent chain drive, countershaft and rope drive	$= 5.4$ h.p.		
Estimated loss in turning head shaft at 700 feet per minute belt speed	$= 2.4$ h.p.		
Total losses up to elevator belt		14.2 h.p.	
Power delivered to elevator belt			93 h.p.
Horse-power to lift 13,000 bushels wheat at 60 pounds per bushel to height of 216 feet		85 h.p.	
Estimated loss at foot shaft.		1 h.p.	
Horse-power due to load and foot shaft			86 h.p.
Horse-power to drag buckets through grain in boot $= B =$			7 h.p.

Torsional pull delivered to elevator belt $= \dfrac{93 \text{ h.p.} \times 33{,}000}{700}$ $= 4400$ lbs.

$$\left.\begin{array}{ll}\text{Weight of belt on each side} & = 216 \times 6.07 = 1310 \text{ lbs.} \\ \text{Weight of empty buckets each side} & = 345 \times 4 \quad = 1380 \text{ lbs.}\end{array}\right\} \quad 2700 \text{ lbs.}$$

$T_1 =$ total tension in up belt (no take-up tension) $= 7100$ lbs.

$T_2 =$ total tension in down belt (no take-up tension) $= 2700$ lbs.

In the example above, if we assume $f = .25$ and $\dfrac{T_1}{T_2} = 2.19$, then for a tension T_2 in the down belt of 2700 pounds, the head pulley would exert a pull

T_1 of $2700 \times 2.19 = 5900$ pounds in the up belt. This is far short of the actual pull of 7100 pounds and it is, therefore, probable that the elevator could not have been driven with a bare iron head pulley unless the boot shaft were loaded or screwed down to put an extra tension in the belt. If the boot shaft were loaded with 2500 pounds, 1250 pounds added to T_1 and T_2 would make them, respectively, 8350 and 3950 pounds; then, $\frac{T_1}{T_2} = 2.12$ and the plain iron pulley might have driven the belt.

The head pulley of this elevator was actually covered with rubber lagging; assuming $f = .35$, then if $T \ = 2700$ pounds, $T_1 = 2700 \times 3.00 = 8100$ pounds. This is an excess of 1000 pounds over the actual pull in the belt or a margin of $\frac{1000}{4400} = 22$ per cent above the work of digging and elevating the grain.

Since the foot of this elevator was loaded with about 1000 pounds by a weighted take-up the normal values of T_1 and T_2 in operation were probably about 7600 pounds and 3200 pounds, respectively. If we say that $f = .35$, and the ratio of tensions corresponding to that value is 3.00, then the pulley would exert a belt pull of $3200 \times 3.00 = 9600$ pounds, or an excess of 2000 pounds over the normal value of T_1 for such contingencies as overload in the boot, loss of driving contact due to dust in the casing, etc. If f should fall to .30, with 3200 pounds on the down side, the pulley would drive the belt with a force of 8200 pounds, and there would be a margin of 600 pounds above the normal value of T_1; if f fell to .27, the pulley would exert a pull of 7600 pounds in the up belt, just enough to drive it.

Value of Take-up Tension.—When a grain elevator 200 feet high, capacity 12,000 bushels per hour, is fitted with an automatic weighted take-up boot, the boot shaft may move vertically 2 or 3 inches during operation between no load and full load with a good new belt. This shows that the elastic stretch may be 1 or 2 inches per hundred feet in such service, and it also indicates that in an elevator equipped with take-up screws, it is not possible to maintain a fixed minimum tension T_2 in the down belt, when part of that load is added, as is usual, at the foot. But when T_2 is kept constant by a weighted or loaded take-up, dust in the casing, or the condition of the belt, or of the lagging on the pulley rim may cause f to vary within rather wide limits without affecting the certainty of the drive. If T_2 decreases, T_1 diminishes also, but in greater amount, because the working ratio $\frac{T_1}{T_2}$ is between 2 and 3. When T_1 diminishes, the digging power of the elevator or its lifting capacity falls off, and it may choke. Hence, in hard-worked elevators, and especially in those which are wet or dusty, it is important to maintain the take-up tension, preferably by weighting the boot shaft.

Calculations of Belt Tensions and Horse-power.—The various items which enter into these calculations have been set down in Table 52; the following comment will explain what they are and how they should be used.

TABLE 52.—CALCULATION OF ELEVATOR BELT STRESSES AND HORSE-POWERS

Item	Stresses in Up Belt at Head Pulley	Stresses in Down Belt at Head Pulley	Horse-power Required
Lifting the load	(1) Weight of material in buckets	(2) Zero	(3) $\frac{\text{Item 1} \times \text{belt speed,}}{33{,}000}$ or $\frac{\text{lbs. per minute} \times \text{height (feet)}}{33{,}000}$
Weight of empty belt and buckets	(4) Add for this	(5) Add for this	(6) Affects Item (15)
Digging the load or filling buckets	(7) Add as a percentage of height of lift. (See page 273)	(8) Zero	(9) Add as a percentage of height of lift. (See page 273)
Frict losses at foot-shaft	(10) Add a small percentage	(11) Zero	(12) Add a small percentage
Friction losses at head-shaft and in power transmission	(13) Zero	(14) Zero	(15) Estimate and add
Air resistance	(16) Generally very small	(17) Zero	(18) Add a small percentage for high-speed elevators only
If belt operates with little or no take-up tension, total pull in belt equals	(19) Sum of Items 1+4+7+10+16	(20) Item 5	(21) Total h.p. = sum of Items 3+9+12+15+18
If added tension is needed to make head-pulley drive	(22) Add enough to make proper ratio $\frac{T_1}{T_2}$	(23) Same as Item 22	(24) Affects 12 and 15 only as friction losses
If belt operates under added tension, total pull in belt equals	(25) Sum of Items 1+4+7+10+16+22	(26) Sum of Items 5+23	(27) Same as Item 21 plus a small per cent for greater friction losses

Item 1. Pull due to Weight of Material.—This equals

$$\frac{\text{pounds of material in 1 bucket} \times \text{height of elevator (feet)}}{\text{spacing of buckets (feet)}}$$

or, what comes to the same thing, it equals

$$\frac{\text{capacity of elevator (pounds per minute)} \times \text{height of elevator (feet)}}{\text{belt speed (feet per minute)}}$$

Item 3. Horse-power to Lift the Load.—These two expressions come from multiplying Item 1 by

$$\frac{\text{belt speed (feet per minute)}}{33{,}000}$$

to convert pounds of pull to horse-power.

Item 4. Pull due to Weight of Belt and Buckets.—The weight of ordinary elevator belt in pounds per linear foot is approximately width (inches) $\times$ number of plies $\times .03$. For accurate weights, see Tables 4, 5, 6, 7, 8. Weights of malleable-iron buckets are given in Tables 39, 40, 41, 42. Weights of other buckets can be taken from manufacturers' catalogues.

Item 7. Pull due to Pick-up.—This can be determined only by experience or experiment. An empirical rule based on long practice is that the pull required to pick up coal, ashes, ores, stone and similar coarse materials from a take-up boot at the speeds of Table 36 is about equal to the weight of the material carried on $12D$ feet of belt, where D is the diameter of the foot wheel in *feet*. For example, if an ore elevator has a 2-foot diameter foot wheel, the work of pick-up is equivalent to adding 24 feet to the height of the elevator.

Experiments by Hanffstengel (Förderung von Massengütern, Vol. 1, 1915) with a short elevator inclined at 30° from the vertical, having a fixed bearing foot with a semi-circular bottom and a chute entering at the level of the foot shaft, gave the results shown in Table 53. The work of pick-up, stated as foot-pounds per pound of material elevated, is equivalent to, and may be expressed as, feet added to the height of the elevator, because the work (foot-pounds) of elevating the material is the work of lifting the same weight of material through the height of the elevator. The figures of the table are lower than those given by the empirical rule stated above, but it is probable that the rule is better suited to ordinary practice than the results derived from the laboratory test.

Table 53 shows that with lumpy material the power required for the pick-up was less when the clearance between the bucket and the boot bottom was small, but where the material was fine, like soft coal slack, the small clearance was of no advantage. This points to what is already known from practical experience, that is, that power is saved in elevators for lumpy material if the bottom clearance is less than the least dimension of the

TABLE 53.—WORK OF PICK-UP IN ELEVATOR BOOT

(Adapted from Hanffstengel)

Material Elevated	Bottom Clearance in Boot, Inches	Work of Pick-up in Foot-Pounds per Pound of Material	
		120 Buckets per Minute	60 Buckets per Minute
Soft coal slack	$2\frac{3}{4}$	4	4.6
Soft coal slack	$\frac{3}{4}$	more than 4	more than 4.6
Boiler house coal (crushed)	$2\frac{3}{4}$	4.2	8.2
Boiler house coal	$\frac{1}{4}$	3.3	5.0
Coke (size not stated)	$2\frac{3}{4}$	5.0	10.0
Coke (size not stated)	$\frac{1}{4}$	3.0	5.0

pieces handled, so that they cannot wedge under the bucket. On the other hand, however, small clearance cannot be maintained in a take-up boot, and it is often inconvenient to use a fixed bearing boot or a special boot (see p. 286). Small clearance leads to greater wear on the bottom sheet of the boot and the risk that the sheet may be cut if the buckets are knocked out of shape or if the foot shaft settles from wear in the bearings. For these reasons, it is better to use the empirical rule in estimating the work of pick-up.

In high-speed grain elevators with foot pulleys one-third or one-fourth the size of the head wheel the grain piles up on the lifting side and the pull on the belt due to pick-up may be equivalent to $10D$ expressed in feet of height, where D is the diameter of the foot pulley in feet. That is, if such an elevator, has a 72-inch head pulley and a 24-inch foot pulley, the pick-up is equivalent to adding 20 feet to the elevator,

With relatively larger foot pulleys and speeds according to Table 36 the work of pick-up for fine dry free-flowing material may be taken as $6D$.

In inclined elevators with spaced buckets run at the speeds of Table 37 the pick-up is easier and the pull may be taken at $4D$ instead of $6D$.

In continuous bucket elevators run at speeds not over 150 feet per minute and properly loaded from a chute the shock or impact from loading is not great and the added pull in the belt due to it may be taken at $2D$ or $3D$. But if the feed is poor and if the spilled material is allowed to accumulate under the foot wheel the pull may be $6D$ or even more. The same is true if the buckets at the loading point are confined within steel plates forming a stationary " loading leg," in which case material catching between the buckets and the plates adds to the pull on the belt.

Item 10. Pull due to Friction Loss at Foot.—One or 2 per cent of the total calculated pull in the elevator belt should be enough to cover this.

Item 15. Pull due to Power Transmission Loss.—Allow 5 per cent for each speed reduction from the source of power through belts, chains or cut gears, and 10 per cent for each reduction through cast gearing.

Item 18. Allowance for Air Resistance.—Five per cent should cover this in high-speed grain elevators.

Item 19. Proper Ratios of Belt Tensions.—If the elevator has a belt wrap of 180° on a plain iron pulley the head pulley will drive, without tightening the belt, if T_1, which is Item 19, is not more than 2.19 times T_2, which is Item 20. If the pulley is rubber-covered, then Item 19 can be 3.00 times Item 20 without excessive slip (see Table 20, page 109).

Item 21. Total Horse-power.—This is also equal to

$$\frac{(\text{Item } 19 - \text{Item } 20)}{33{,}000} \times \text{belt speed (feet per minute)}$$

plus the allowances added in Item 15.

Item 22. Calculation of Added Take-up Tension.—Suppose $T_1 =$ Item $19 = 2000$ pounds and $T_2 =$ Item $20 = 800$ pounds, then the ratio is $\frac{2000}{800} = 2.5$; it is greater than 2.19 and a plain iron pulley will not drive the belt unless extra tension is added. To find what tension x must be added to each side of the belt to make $T_1 = 2.19\,T_2$, we say $2000 + x = 2.19\,(800 + x)$, from which $x = 210$ pounds, which is Item 22.

Item 23. Added Belt Tension due to Take-up.—The total tension to be added by the take-up is 420 pounds, divided between the up belt and the down belt.

Maximum Belt Tension.—The total pull for which the belt should be selected is either Item 19 or Item 25.

If we assume a unit tension p, then for a belt of width W, the number of plies $= \dfrac{\text{maximum tension (Item 19 or 25)}}{pW}$.

Working Tensions for Belts.—What is said on page 111 about unit stresses applies generally to elevator belts as well as conveyor belts. In most cases the stress can be kept below 25 pounds per inch per ply for 32-ounce duck; but there are many successful elevators in which the unit stress is 30 pounds. It should not exceed 35 pounds for 32-ounce duck, nor 40 pounds for 36-ounce duck.

Thickness of Belts as Determined by Wear.—Elevator belts for coarse materials are usually made some plies thicker than is necessary to transmit the maximum tension, because in most cases the life of the belt is determined by the external wear and damage it receives (see p. 249). An extra thickness acts in such cases as a protection against belt failure by reason of loss of tensile strength; it also gives the belt strength and stiffness to back up the bucket and prevent the bucket bolts from pulling through under severe strain. This is shown by some of the elevators listed on page 238. In Elevator 1 the daily tonnage would, if evenly spread over every minute of the twenty-four hours, require each bucket to be loaded only one-fourth full. If it is assumed that the buckets are at times three-fourths full, the maximum tension corresponding to Item 25 is 7500 pounds, and if the belt

is stressed to 26 pounds per inch per ply, 8 plies are required. Actually 11 are used. In Elevator 2 the maximum tension is 8100 pounds, requiring less than 9 plies, but the belt has 12. Similarly in Elevator 4, the maximum tension corresponding to a load three times the average, based on tonnage, plus the water carried, is 6800 pounds; this requires less than 9 plies, but the belt has 12 plies.

Minimum Number of Plies in Belt.—Practical experience has shown that to give good service belts should have a certain minimum number of plies, based on the considerations stated above, regardless of the tensile strength required. Table 54 is a fair statement of modern practice.

TABLE 54.—MINIMUM NUMBER OF PLIES IN ELEVATOR BELTS

Material Elevated	Width of Belt, Inches		
	10 to 18	20 to 28	30 and over
Grain, flour-mill products, chips, bark, materials not abrasive	4	5	6
Fine material, 50–75 lbs. per cu. ft., materials not too coarse	5	6	7
Heavy, coarse materials, ore, crushed stone, lump coal.	6	7	8

Belt Slip.—When a belt slips on the head pulley there is a reduction of elevating capacity and the danger of a choked boot if the feed continues at the normal rate. In this case the belt may slow down, pull some buckets off and eventually stop. Even though slip may not come to the point at which the belt stops, wear on the pulley side of the belt is sure to follow. In a rubber belt the duck on one side may be frayed or worn off. In a stitched canvas belt the same thing happens or the rubbing glazes the surface on the pulley side of the belt if it has been painted, and if it has been impregnated with a Class 1 compound (see p. 47) the heat generated by the rubbing is apt to dry up the compound and make the belt brittle. This is not true of Class 2 compounds, because they resist the action of heat much better.

The rubber covering of head pulleys is subject to the same wear as the belt, but since the lagging exposes less surface to it than the belt, the damage is apt to be greater. The heads of the bucket bolts often tear the lagging when slip occurs, and with that is combined the wear due to belt creep (see p. 276).

Fig. 238 shows bolts used to fasten rubber lagging to the rim of a grain-elevator head pulley. Some are bent by the pull on the lagging, and the heads are worn off by the slip and creep of the elevator belt. Wearing away the metal causes damage to the pulley side of the belt which adds to that caused by slip and creep.

If the choke is so bad that the belt stops, it is important that the pulley should stop also. In modern elevators with separate electric motor drive

an overload release device can be used to throw off current and stop the motor if the load becomes too great; but in old-time elevators, driven from line shafts, it has often happened that a head pulley continued to turn when the belt was stopped by a choke and the belt was worn in two by the pulley and fell down the legs. In some cases the damage was more serious; the friction from the revolving pulley set fire to the belt, the flame caused a dust explosion in the elevator casing and from that followed loss of life and a fire which caused the total destruction of the building and its contents (see p. 302).

Belt Creep; Belt Slip.—When an elevator belt is so heavily loaded that the head pulley will not drive it the belt may stand still while the pulley turns within the loop. The slip in such a case may be called 100 per cent of belt travel, because the travel of the belt has fallen to zero. If the load is less, the slip is less, but it is always there to some degree, although it may amount to only a fraction of 1 per cent of belt travel when the load is light (see also Chapter V).

Belt creep is different. In Fig. 250, representing the head of an elevator, the up belt under tension T_1 assumes a certain stretch, so that a section of belt normally 12 inches long under no load may be pulled to $12\frac{1}{8}$ inches long just as it reaches the pulley. A second or two later that section of belt has reached the other side of the pulley, the tension has fallen to T_2 and the length is only $12\frac{1}{16}$ inches—that is, in passing over the pulley, every foot of elevator belt shortens $\frac{1}{16}$ inch, and if the belt speed is 640 feet per minute, the creep or relative movement between the belt and the rim of the pulley is 40 inches per minute or 114 miles per year of average service.

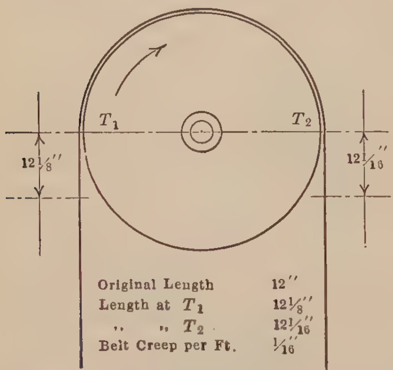


Fig. 250.—Creep at Head of Belt Elevator.

An illustration of belt creep suggested by W. W. Bird (Transactions A. S. M. E., 1905) is to stretch an ordinary elastic band as a belt over two small pulleys of equal size. If the band has any resistance to overcome, the driven pulley may turn only half as fast as the driver. In this case the 50 per cent loss in transmission is not due to slip at all, it is due to the creep of the highly elastic band on the pulleys.

The relations between slip and creep and load, as observed in some of Wilfred Lewis's experiments, are given in Table 50 (see p. 267). Column 5 states the combined slip and creep in per cent of belt travel as registered by the apparatus used in the tests. In Column 6 Barth has calculated the creep alone from the observed stretch of leather belts under load. In Lewis's experiments no tests were made of rubber or other fabric belts, but so far as creep and slip are concerned, it is probable that the differences

between leather belts and fabric belts would be in degree only, and not in manner. Assuming that the differences are not great, we can take an example from Tests 128 and 131, Table 50. Here the ratios of $\frac{T_1}{T_2}$ are between 2 and 3, as in general elevator practice, and the tension in the belt, 511 or 557 pounds per square inch, corresponds to between 30 and 35 pounds per inch per ply of fabric, which is not unusual in elevator belts. Here the combined slip and creep (see Column 5) amounts to 0.5 to 1.1 per cent of belt travel, and by calculation practically all of it is creep. It may be said then that in elevator practice a creep of at least ½ of 1 per cent is unavoidable.

If the load on the elevator belt is increased, T_1 would increase and the belt might slip, unless T_2 were increased by putting more tension on the belt by screwing down or loading the boot shaft. Then with a clean belt, the coefficient of driving contact might rise slightly (see Experiment 133, Column 7), and the head pulley would drive the belt, but with 0.5 per cent slip plus 1.3 per cent creep. This sum represents a loss of nearly 2 per cent of belt travel; it shows what may happen in an elevator severely overloaded.

Effect of Creep and Slip.—It is doubtful whether the coefficient of belt friction in the instance mentioned above would increase in the same proportion in a wet or dusty elevator; but in any case, the slip is objectionable. It means not only a loss of elevator capacity, but, what is more injurious, a movement between the pulley rim and the belt, which, under the pressure due to the load at the head of the elevator, tends to wear away the pulley side of the belt, or the lagging of the pulley, if it is rubber-covered.

Lagging on head pulleys does not last as long as the elevator belt; the two materials are usually of the same class of belting, but the wear on the belt is spread over perhaps 100 to 400 feet of length while the wear on the pulley covering is confined to 10 to 25 feet of lagging. Besides, the lagging is often worn and torn by the heads of the bucket bolts projecting beyond the pulley side of the belt, especially when the bolts are loose or the belt worn. The wear is also greater when the elevator belt is of poor quality or too thin for the work; a belt that stretches greatly under load will creep more on the head pulley than a firm belt, not overloaded, from which the excess elasticity of the fabric has been removed during the process of making the belt.

Pulleys with Rough Rims, or rims not turned, or rims cast with depressions or slots parallel to the shaft have been used in elevators to increase the grip on the belt. It is doubtful whether the coefficient of belt contact on such a pulley is any greater than on a smooth rim, but there is no doubt that when a belt creeps, as it must, or when it slips, as it may, it is sure to be damaged by the rough rim. If the elevator is overloaded, so that the belt stops while the pulley continues to turn, a rough rim may ruin a belt in a few minutes. The same thing may happen if the elevator is started under load and with a choked boot, as a result of an enforced shut-down

caused by failure of electric power, or by some mishap to the elevator machinery or its accessories.

In the Western mining country, belt elevators are in general use for handling pulps and wet ores in the process of concentrating. When the mixture is largely water, the spill and splash at the head and the material in the boot keep the belt and pulleys wet, the belt slips on the head pulley and is worn on the pulley side. Experiments have been made with devices to prevent slip, such as slatted-faced pulleys, rough-rim pulleys, etc., but they add to the wear on the belt for the reasons stated above. The real cure for slip is a strong belt, more tension in the belt and, if possible, the use of an automatic foot take-up to maintain that tension. On this point, see page 299.

Patents have been issued on pulleys with projecting cleats or buttons and on pulley lagging with a roughened face. None of these devices would work at the head of an elevator. The belt must creep and it may slip; any bodily interference with these natural movements will injure the belt.

Elevator Head Pulleys are sometimes made split, that is, in halves, for convenience in handling or getting on or off the shaft; but when they are made in the ordinary way with the heads and nuts of the clamping bolts bearing against rough unfinished surfaces the bolts are apt to work loose under vibration and heavy loads. For important work, split pulleys should have finished bearing pads around the bolt holes at hub and rim. Some engineers will not use split pulleys at all, but specify clamp hub pulleys fitted with not over .002-inch clearance to a shaft turned to accurate size. When standard grade pulleys are put on cold rolled or turned shafting of commercial grade the clearance may be .005 inch or more, and very often the pulley works loose and shifts on the shaft in spite of keys and set screws. Some engineers have their head pulleys bored a few thousandths smaller than the accurately turned shaft and then pressed or driven on; others use clamp hub pulleys, heat the hub bolts nearly to redness, put them in place, tighten the nuts and let the contraction of the bolts clamp the hub even tighter and prevent the nuts from loosening. In some large ore elevators (Ohio Copper Co., Lark, Utah, Eng. and Min. Jour., Vol. 99) the head pulleys are 60 by 38 inches, split, bored $5\frac{7}{16}$ inches with 2 keys at 90° apart, and the outside of the hub at each end is turned to take a $\frac{5}{8}$ by 5-inch welded steel band shrunk on. These pulleys stay tight; to remove them, the bands must be cut apart.

Lagging consists usually of 3- or 4-ply rubber belt fastened on by $\frac{1}{4}$-inch flat-head bolts. It is important that the bolt heads should be below the surface of the lagging; for this reason the pulley rim should be countersunk around the bolt hole to allow the belt to be depressed (see Fig. 104, p. 123). Pulley faces up to 18 or 24 inches can be covered by a strip of belt of that width; for wider faces and heavy crowns it is easier to use two strips of belt of half the width.

An improved pulley lagging consists of 2- or 3-ply rubber belt with a cover $\frac{1}{16}$ or $\frac{1}{8}$ inch thick. This does not cost more than 4-ply belt, it resists

abrasion better and is not so likely to be injured by projecting heads of bucket bolts (Fig. 238). On the wear of lagging, see page 250.

Face, or Width, of Head Pulleys is usually 1 or 2 inches more than belt width, up to 30 or 36 inches and 3 or 4 inches more for wider belts. The excess of face permits the belt to run out of center for an inch or two, in case the head shaft is not leveled properly or gets out of level by shrinkage or settling of supports.

The crown of standard transmission pulleys is $\frac{1}{8}$ inch on the diameter per foot of face—that is, the face of a pulley for a 12-inch belt is $\frac{1}{16}$ inch higher in the middle than at the edges. In most cases, elevator head and foot pulleys are made in the same way, but some engineers prefer a crown of $\frac{3}{16}$ inch per foot or even $\frac{1}{4}$ inch per foot for the head pulleys of elevators that are hard to keep in alignment. When the belts are wide and the buckets are in single row the extra high crown may be objectionable (see p. 261),

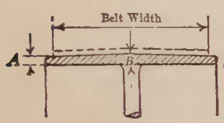


TABLE 55.—RIM THICKNESS—INCHES, OF STANDARD D. B. PULLEYS

Diameter, Inches	Width of Belt—Inches										
	12	14	16	18	20	22	24	28	32	36	40
30	$A=\frac{9}{32}$ $B=\frac{3}{8}$	$A=\frac{9}{32}$ $B=\frac{3}{8}$	$A=\frac{9}{32}$ $B=\frac{3}{8}$								
36	$A=\frac{9}{32}$ $B=\frac{3}{8}$	$A=\frac{9}{32}$ $B=\frac{3}{8}$	$A=\frac{9}{32}$ $B=\frac{3}{8}$	$A=\frac{13}{32}$ $B=\frac{1}{2}$							
42	$A=\frac{11}{32}$ $B=\frac{7}{16}$	$A=\frac{11}{32}$ $B=\frac{7}{16}$	$A=\frac{11}{32}$ $B=\frac{7}{16}$	$A=\frac{13}{32}$ $B=\frac{1}{2}$	$A=\frac{3}{8}$ $B=\frac{1}{2}$						
48	$A=\frac{13}{32}$ $B=\frac{1}{2}$	$A=\frac{13}{32}$ $B=\frac{1}{2}$	$A=\frac{13}{32}$ $B=\frac{1}{2}$	$A=\frac{7}{16}$ $B=\frac{17}{32}$	$A=\frac{13}{32}$ $B=\frac{17}{32}$	$A=\frac{13}{32}$ $B=\frac{17}{32}$					
54	$A=\frac{15}{32}$ $B=\frac{9}{16}$	$A=\frac{15}{32}$ $B=\frac{9}{16}$	$A=\frac{15}{32}$ $B=\frac{9}{16}$	$A=\frac{15}{32}$ $B=\frac{9}{16}$	$A=\frac{15}{32}$ $B=\frac{9}{16}$	$A=\frac{1}{2}$ $B=\frac{5}{8}$	$A=\frac{1}{2}$ $B=\frac{5}{8}$				
60	$A=\frac{15}{32}$ $B=\frac{9}{16}$	$A=\frac{15}{32}$ $B=\frac{9}{16}$	$A=\frac{15}{32}$ $B=\frac{9}{16}$	$A=\frac{15}{32}$ $B=\frac{9}{16}$	$A=\frac{15}{32}$ $B=\frac{9}{16}$	$A=\frac{1}{2}$ $B=\frac{5}{8}$	$A=\frac{1}{2}$ $B=\frac{5}{8}$	$A=\frac{19}{32}$ $B=\frac{11}{16}$	$A=\frac{21}{32}$ $B=\frac{3}{4}$		
66	$A=\frac{17}{32}$ $B=\frac{5}{8}$	$A=\frac{17}{32}$ $B=\frac{5}{8}$	$A=\frac{17}{32}$ $B=\frac{5}{8}$	$A=\frac{17}{32}$ $B=\frac{5}{8}$	$A=\frac{17}{32}$ $B=\frac{5}{8}$	$A=\frac{5}{8}$ $B=\frac{3}{4}$	$A=\frac{5}{8}$ $B=\frac{3}{4}$	$A=\frac{23}{32}$ $B=\frac{13}{16}$	$A=\frac{25}{32}$ $B=\frac{7}{8}$		
72	$A=\frac{19}{32}$ $B=\frac{11}{16}$	$A=\frac{19}{32}$ $B=\frac{11}{16}$	$A=\frac{19}{32}$ $B=\frac{11}{16}$	$A=\frac{19}{32}$ $B=\frac{11}{16}$	$A=\frac{19}{32}$ $B=\frac{11}{16}$	$A=\frac{11}{16}$ $B=\frac{13}{16}$	$A=\frac{11}{16}$ $B=\frac{13}{16}$	$A=\frac{25}{32}$ $B=\frac{7}{8}$	$A=\frac{25}{32}$ $B=\frac{7}{8}$	$A=\frac{27}{32}$ $B=\frac{15}{16}$	
84	$A=\frac{21}{32}$ $B=\frac{3}{4}$	$A=\frac{21}{32}$ $B=\frac{3}{4}$	$A=\frac{23}{32}$ $B=\frac{13}{16}$	$A=\frac{23}{32}$ $B=\frac{13}{16}$	$A=\frac{11}{16}$ $B=\frac{13}{16}$	$A=\frac{11}{16}$ $B=\frac{13}{16}$	$A=\frac{11}{16}$ $B=\frac{13}{16}$	$A=\frac{25}{32}$ $B=\frac{7}{8}$	$A=\frac{25}{32}$ $B=\frac{7}{8}$	$A=\frac{27}{32}$ $B=\frac{15}{16}$	$A=\frac{13}{16}$ $B=\frac{15}{16}$
96	$A=\frac{23}{32}$ $B=\frac{13}{16}$	$A=\frac{13}{32}$ $B=\frac{13}{16}$	$A=\frac{23}{32}$ $B=\frac{13}{16}$	$A=\frac{23}{32}$ $B=\frac{13}{16}$	$A=\frac{3}{4}$ $B=\frac{7}{8}$	$A=\frac{3}{4}$ $B=\frac{7}{8}$	$A=\frac{3}{4}$ $B=\frac{7}{8}$	$A=\frac{27}{32}$ $B=\frac{15}{16}$	$A=\frac{27}{32}$ $B=\frac{15}{16}$	$A=\frac{29}{32}$ $B=1$	$A=\frac{7}{8}$ $B=1$

but this is not so when the buckets are bolted to the belt in two rows (Fig. 244).

Flanged pulleys should never be used on elevators, either at head or foot. Contact between the flanges and the edges of the belt will cut or wear off the edges.

Rims of Head Pulleys are not turned to exact widths or thicknesses, and the dimensions vary with different makers. Table 55 gives approximate thicknesses of double-belt pulleys at the edge and center of rim for the sizes generally used at the head of centrifugal discharge elevators. For high elevators handling grain and for large elevators for minerals and ores it is advisable to use pulleys with thicker rims and heavier arms. A thicker rim is also recommended for cases where the pulley is not lagged and the belt handles wet, gritty material. The slip and creep of the belt combined with the grit which sticks to the belt often wears out the rim of a standard pulley. For creep and slip, see page 276.

CHAPTER XXI

ELEVATOR BOOTS

Purposes of Elevator Boots.—An elevator boot serves several purposes:

1. To confine the material to the path of the buckets;
2. To support the foot shaft;
3. Generally, but not always, to support the elevator casing.

There are two general types of elevator boots—boots with fixed shaft bearings and boots with take-up bearings.

Fixed Bearing Boots.—Where elevator pits are small, or hard to get at, or blocked at times by spill from feeding conveyors, or by chokes in the elevator itself, it may be an advantage to use boots with fixed bearings and put the take-up bearings at the head of the elevator. This is standard practice in some cement mills; the elevators are provided with stairs and platforms which make it convenient to reach the head to adjust the take-ups. The take-up screws in this position are comparatively clean, but in a deep pit they become dirty and hard to turn in spite of dust guards and ordinary efforts to keep the pit clean. In some mills, pits are never permitted to get dirty with spilled material; but in others, with the class of labor employed, the great volume of material handled, and the twenty-four-hour continuous operation, it is practically impossible to keep them clean at all times.

Fig. 251 shows a pit for the foot of an elevator for crushed cement rock $\frac{1}{4}$ inch and under. The entire structure below the floor line is concrete, the foot bearings rest in openings in the 8-inch walls forming the sides of the boot. Sheet-steel doors on each side give access to the interior of the boot and a chute with a slide door permits spilled material or material which has been cleaned out of the boot to be shoveled from the pit into the path of the buckets.

There are other reasons for using boots with fixed bearings. In a take-up boot, with the wheel in its upper position, there is a mass of material lying beyond the sweep of the buckets which may pack so hard that unless it is dug loose by hand before the wheel is set down to a new position the buckets will not pick it up, but will be torn loose or pulled out of shape. This is true of some chemicals and fertilizers, and of cement; for these, a boot with a definite sweep of the buckets is best. Fixed bearing boots are also used for food products which spoil if allowed to accumulate beyond the sweep of the buckets, also in elevators which at different times handle different materials which should not be mixed.

If the material handled is coarse, hard and lumpy there is some advantage in using a fixed-bearing boot, so that the pieces pushed by the buckets slide over a curved steel bottom plate at a fixed distance from the foot shaft and do not drag over a bed of lumps lying beyond the sweep of the buckets (see p. 273 and Table 53). There is, however, a limitation to this; if the material is approximately round or cubical in shape, buckets can be run close to a bottom sheet, but if the pieces are sharp-cornered, or angular or long, like slivers, there is danger that they may wedge between the buckets and the bottom sheet and either damage the belt or chain or tear the bucket loose or punch a hole in the sheet. If these materials are handled in a centrifugal discharge elevator, which is not always the best way, then the clearance should be larger than the dimension of the lump and the bucket should be large enough and strong enough and the belt or chain that carries it sufficiently powerful to dig the material as if from its own bed.

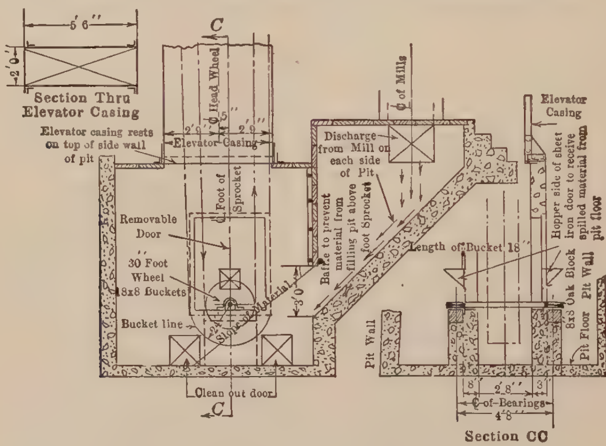


FIG. 251.—Concrete Pit and Boot for Pulverized Stone Elevator.

In many elevators for sand, ashes, coke and similar abrasive substances the boot is made as the lower part of the elevator casing and not as a separate piece; in these elevators it is not customary to use a curved-bottom sheet, nor even a flat-bottom sheet; the floor on which the casing rests forms the bottom. This construction saves something in first cost and also in renewals, for a curved boot bottom does not last long when the elevator handles lumpy and abrasive materials.

In some elevators even less attempt is made to confine the material to the path of the buckets; in hot-clinker elevators, for instance, as used in cement mills, the boot consists of a trench or pit with the shaft bearings mounted on the side walls or on cast-iron pedestals between the walls. The clinker then forms its own boot under and alongside the foot wheel. Most of it is like sand or gravel in size, and even if a large fused mass of clinker or a portion of dislodged kiln-lining too large for the buckets should slide down into the pit, it will not jam there, as the buckets can tumble it

out of the way. These clinker elevators are chain machines, but the arrangement described is applicable to some belt elevators as well.

Manufacturers' designs of fixed bearing boots are of steel plate throughout or with cast-iron sides and a steel-plate bottom. Fig. 252 shows one which can also be furnished with a sectional cast-iron bottom to take the wear from abrasive materials. Fig. 253 shows a boot for fine crushed rock, designed by a cement company for its own use; it is fed at front and back. The $\frac{1}{4}$-inch bottom sheet is not fastened in place, but is held in a $\frac{5}{16}$-inch slot between a bent angle and a bent flat fastened to each side of the boot. The shaft bearings can be easily removed, a retainer ring on the inside of the boot holds the heads of the bolts from turning and keeps them from falling out of place.

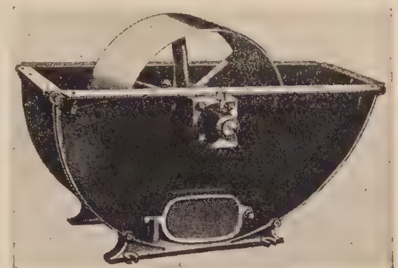


Fig. 252.—Cast-iron Fixed-bearing Boot. (Jeffrey Mfg. Co.)

Designers of elevators sometimes choose fixed bearing boots so that the elevators can be driven at the foot without interfering with the take-up. This is perhaps the poorest reason for using such boots. Belt elevators carrying light stuff, like wood chips, have been driven at the foot, but the head take-ups need constant attention to keep the belt in driving contact with the foot pulley. In chain elevators driven at the foot the chains slip and break unless they are kept at a steady tension and free from slack. Experienced engineers do not drive elevators at the foot.

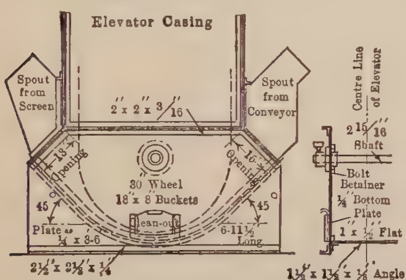


Fig. 253.—Double-sided Boot for Pulverized Cement Rock.

Take-up Boots.—Over thirty styles of take-up boots are sold by American manufacturers, and most of them are made in a number of sizes. There are two principal kinds—boots for coal and similar coarse, heavy substances and boots for grain. The distinguishing features of these boots are determined by the pick-up of material; for a discussion of this, see pages 217, 219.

Boots for Coal and similar coarse, heavy materials consist of a pair of sides, usually cast iron, joined by a curved bottom plate, usually of steel plate. The front is sloped to direct material into the path of the buckets and at the same time to form a clearance space into which the pieces may be pushed instead of jamming against the bottom sheet, as might happen if the bottom were fully rounded with the lower position of the shaft as a center.

Fig. 254 shows a section through one side of a standard boot. The bearings, closed at the outer end to prevent loss of lubricant, are made with a spherical center so that the shaft will not be cramped even if the two bearings are not adjusted alike, or if the shaft is not square with the sides of the boot. A rectangular slide frame holds the bearing and moves up and down in a slot in the side of the boot; a cap for the slide frame allows the bearing to be removed easily. There is a steel plate on the inside of the boot which travels with the shaft and keeps the slot closed in any position of the shaft. Springs acting on clamp bolts keep the plate tight against the boot side, prevent dust from coming out and keep material away from the bearings and the slides. The screw works in a nut held in a recess in the boot side by a separate yoke piece, a small casting which will break and save the boot side in case of a choke or an accident in the boot. An opening, or clean-out, in each side of the boot, is closed by a cover, recessed to form a dust seal and paneled inwardly so that it comes flush on the inside of the boot and does not leave a cavity or depression in which pieces of material might be jammed and then tear buckets off or damage the belt. This feature is not important in a grain boot, but it is of value in a boot that handles hard, coarse material.

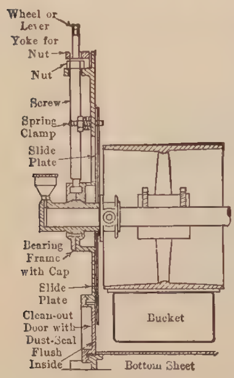


Fig. 254.—Cross-section through One Side of a Boot for Coal and Minerals. (Link-Belt Co.)

Dust-tight Boots.—Other boots, in styles similar to the above, have removable cast-iron bottom plates for greater durability; others, made to be dust-tight, have covers outside the bearings in addition to the inside slide plates (Fig. 255). The jaw-levers pivoted to the top of the take-up screws give a large leverage for adjusting the bearings and do not interfere with the elevator casing as a hand-wheel would.

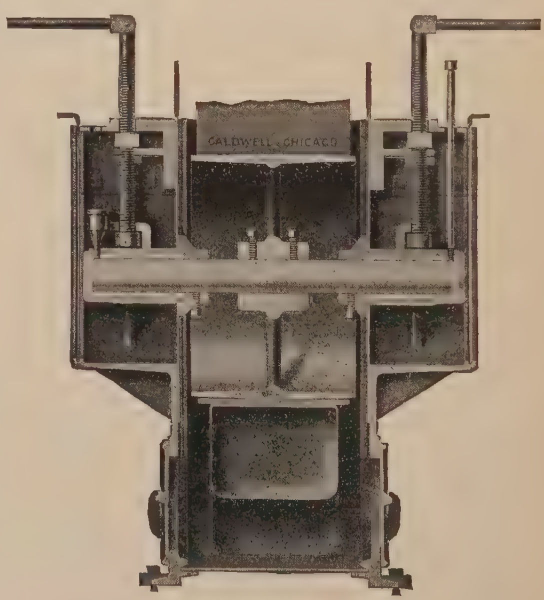


Fig. 255.—Take-up Boot with Outside Dust-seals.

Boot Take-ups.—When boots are made as the base sections of steel or wood elevator casings the take-ups are necessarily separate parts bolted on. Fig. 256 shows one style made with a slide plate to keep the slot in the side of the boot closed at all times.

When a boot works in a damp or wet place it is advisable to make the take-up nuts of brass, so that the take-up screws will not rust tight in them.

Boots in Missouri Lead Mines.—Standard boots listed by manufacturers are well suited to handle coal and other substances which are not too hard or too abrasive. With such materials, the wear on the boot sides and bottoms is not too rapid, but for elevators that carry ores and minerals in ore-reduction plants, concentrator plants and cement mills, the boots are often of special design, with the idea of resisting abrasion by making the sides and bottoms very heavy, or else dispensing with iron

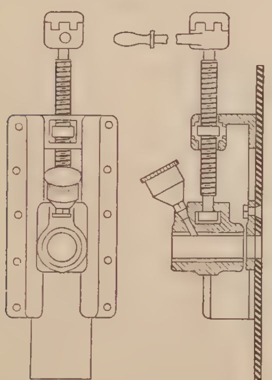


Fig. 256.—Take-up with Dust-seal for Steel or Wood Boot.

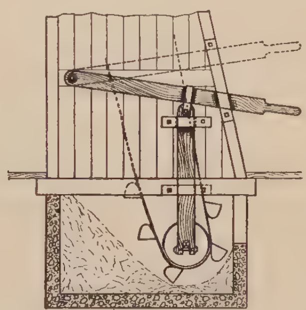


Fig. 257.—Boot for Crushed Ore, Joplin, Missouri.

and steel construction entirely. In the lead-mining district of Missouri crushed ore, in size from sand up to $\frac{3}{8}$ or $\frac{1}{2}$ inch, is handled wet or dry in belt elevators with boots that consist of a rectangular box of planks or a concrete pit which in width has a foot of clearance on each side of the pulley, and in length has room enough for a man to get in and use a shovel behind the pulley. These boots may be 6 or 8 feet below the floor of the mill. The tension is put on the belt and slack taken up, not by screws, but by the arrangement shown in Fig. 257. A 4 by 6-inch vertical post is notched at the bottom end to take the shaft, and the shaft turns in the notch. A pivoted lever, held in position on each side of the wooden casing by wedges or clamp bolts, acts down on the 4 by 6-inch post and applies pressure to the ends of the foot shaft. In some of these elevators the shaft has a flanged sleeve set-screwed on it at each end to take the wear from the pressure-post; when the sleeve wears out it is thrown away and a new one is slipped on. No attempt is made to lubricate it.

Take-up by Moving the Whole Boot.—Fig. 258 shows a boot that has a take-up arrangement and yet maintains a fixed clearance between the sweep of the buckets and the bottom plate. The elevator handled crushed dolomite; it had to be inclined on account of the position of the bin which it fed; and because of the small space available, the casing had to be very narrow with the buckets running on guide angles. It was necessary to prevent an accumulation of material in the boot, and since take-ups could not be placed at the head, the boot was therefore mounted on a frame pivoted at one end and adjusted by two bolts to give the necessary take-up travel.

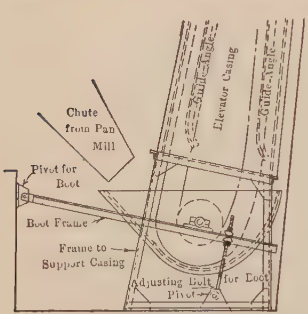


FIG. 258.—Take-up Adjustment by Moving the Whole Boot.

Position of Feed.—From what has been said in Chapter XV it is clear that in a centrifugal discharge elevator the buckets take their load on the rising side of the foot wheel, and at the ordinary speeds of Tables 35 and 36, the buckets cannot take a full load unless they receive some material at or above the level of the foot shaft. This is especially true if the foot wheel is smaller than the head wheel. How this affects the delivery of grain to buckets in a high-speed grain elevator has been shown in Figs. 207 and 208.

When a centrifugal discharge elevator takes coal or some similar material from a sloped-front boot the action is very much the same. Fig. 259 shows a standard form of boot with a front sloped at 45° and with the lower edge of the feed chute about level with the center of the foot wheel in its upper position of take-up travel. When the elevator starts, and material is fed into the boot, little or none of it is caught by the buckets until a bed of dead material forms beyond the sweep of the buckets and piles up in the front of the boot. If the material is fine and dry and stands at a low angle of repose, the bed of dead material will not reach up into the chute; if it stands at 45°, as shown in the figure, it will partly block the chute; if it is damp and sluggish like foundry sand or boiler-house ashes, or bituminous coal or other material received in open cars and exposed to rain or snow, then the angle of repose may be so steep that the chute will be choked and the flow of material stopped.

When the wheel is in the lowest position of take-up travel (Fig. 260) the boot is kept clear of accumulated material, and the disturbance of the material beyond the reach of the buckets on the rising side of the wheel prevents it from blocking the chute.

There are standard forms of take-up boot with steep fronts, the angle being more than 45°. Fig. 261 shows one with a 55° front and with the lower edge of the feed chute *above* the center of the foot wheel in its upper position of take-up travel. When the wheel in such a boot is all the way up, the bed of dead material beyond the sweep of the buckets will not

back up in the chute unless it stands at a steep angle, and hence this boot is likely to work better than a low-front boot if the material is damp, sluggish or apt to pack hard and stand on a steep angle of repose. When the wheel is in its lowest position (Fig. 262) there is even less chance of a choked chute, and the buckets fill well.

One drawback of a steep-front boot is that when the take-up is all the way down, an excess of feed over elevator capacity may cause the foot wheel to be buried so deep that the buckets cannot dig themselves free. This may happen if a heavy load is dumped into the boot or if the elevator should slow down or stop for any reason while the feed continues. The line *A—B* in Figs. 260 and 262 shows how the material may pile in the two styles of boot under such circumstances.

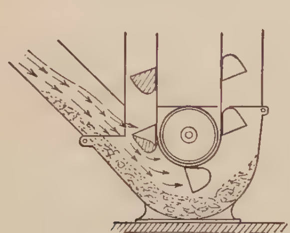


Fig. 259.
45° slope boot, wheel in High Position.

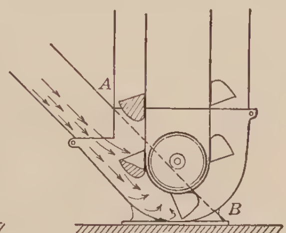


Fig. 260.
45° slope boot, wheel in Low Position.

Fig. 261.
55° slope boot, wheel in High Position.

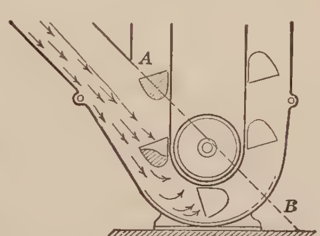


Fig. 262.
55° slope boot, wheel in Low Position.

Summary.—Boots with fronts sloped at 45° (Figs. 259 and 260) have a low chute, save some depth of elevator pit, work well with dry, free-flowing materials, but not so well with sluggish materials unless the foot wheel is set low and the wheel and buckets are of a size that makes the sweep of the buckets come close to the bottom sheet. Boots with fronts sloped at 55°, 57½° or more are higher in front, require some inches more of depth in the boot pit, work well with all materials fit to handle in centrifugal discharge elevators, but require free-flowing substances to be fed with some care, so as not to swamp the foot wheel when it is in a low position of take-up travel (see Figs. 261 and 262).

Fly-feed or Scoop-feed.—In an elevator with continuous buckets run at comparatively slow speeds it is possible to chute the material directly into the buckets, but in centrifugal discharge elevators, where the buckets are spaced apart on a belt or chain and travel at the speeds given in Table 35 or 36, there is little or no gain in an attempt to deliver the material directly to the buckets, the so-called "fly-feed." If the lip of a chute set for "fly-feed" is at or below the level of the foot shaft it is inoperative, because the material will be thrown out of the buckets by centrifugal force (see p. 215) and will drop down and be scooped up when enough has accumulated under and in front of the foot wheel. If the chute is higher and directs the stream of material above the center of the wheel into buckets moving at a speed of 5 to 10 feet per second, only a part of it is caught, most of it falls down into the boot, and some may be splashed and scattered so as to fall on the top of the foot pulley. Material caught between the pulley and the down belt wears or cuts the belt, and with the "fly-feed" the other side of the belt suffers some injury from the direct impact of material striking it between the buckets.

The best feed for hard, gritty, lumpy materials handled in centrifugal discharge elevators is the "scoop-feed" where the buckets act on a yielding mass fed in on the sloping front of a boot, or where the delivery is such that the material can pile to form its own boot. The slope of the front of the boot and the height of the chute with reference to the foot wheel depend on the nature of the material, whether it is free-flowing or sluggish (see p. 287).

Feeding into Side or Back of Boot.—Since buckets in centrifugal discharge elevators never pick up material below or behind the foot wheel, feeding into the boot at those places merely adds to the work of the elevator belt and buckets by the power required to drag the material over the inside surfaces of the boot or over a bed of dead material from the place of feeding to the front of the wheel where the buckets pick up their load. In grain elevators this does no particular harm to the belt or buckets because the grain moves freely and causes very little wear, but in elevators handling lumpy or abrasive material the bottom sheet and side plates of the boot, and also the lips of the buckets, are apt to be worn by the friction of the material dragged around beneath the wheel. A more serious matter is the chance that pieces may wedge fast between the buckets and the bottom sheet, or between the buckets and the bed of dead material lying beyond the sweep of the buckets, with the result that buckets are torn apart or pulled off the belt, and with consequent injury to the belt itself.

It is often convenient to feed an elevator from the back or side to save depth of pit and height and length of feed chutes. When the material is in small pieces this is permissible, but it is dangerous to handle hard and lumpy material in this way.

Where the foot of a grain elevator is lowered into a cargo of grain the foot frame of the "marine leg" must be open at sides and back to feed to the buckets. Fig. 263 shows such an open frame; when it is in operation

the most active flow of material is at the back where the buckets go down.

Size of Foot Pulleys.—It is a common mistake to make these pulleys too small. The object may be to save space, or to avoid a few inches greater depth of pit, or to save something in the price of a boot.

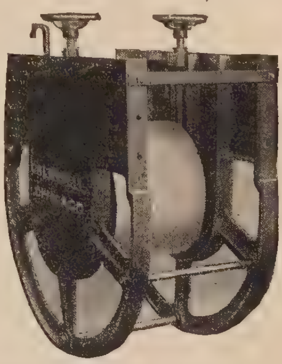


Fig. 263.—Open Boot or Foot Frame of Marine Leg.

So far as the belt is concerned, the general objection to a small pulley is that it tends to separate the plies of fabric in a built-up belt by stretching or breaking the bond which holds the plies of fabric together, whether that bond is stitching, as in canvas belts, or some cementing compound, as in rubber or balata belts, or the binder threads in a solid-woven belt. In conveyor practice, foot pulleys have seldom less than 3 or 4 inches of diameter per ply of belt. Belt elevators are generally shorter between centers than belt conveyors and hence the belt bends over the pulleys oftener for the same belt speed. For that reason, foot pulleys of belt elevators should be at least 4 inches in diameter for each ply of belt—that is, a 24-inch pulley for a 6-ply belt. If the elevator is short, the belt bends oftener, and a ratio of 5 to 1 is better still, to prolong the life of the belt and postpone the day when the belt will fail by separation of the plies. It must be said, however, that most elevator belts are not discarded on account of the internal wear which causes the plies to separate; some belts handling clean, dry substances like grain may die of old age, but most elevator belts succumb to external injuries which have no connection with the diameter of the pulleys they run on.

The great objection to a small foot pulley is the bad pick-up. For reasons stated in Chapter XV and shown in Figs. 204, 206 a small foot pulley may not permit the buckets to pick up any material until they are on the straight run above the pulley and away from the influence of centrifugal force. When that happens, the buckets, while they are in contact with the foot wheel, do nothing but stir up material uselessly, wear themselves out and perhaps injure the belt.

The larger the foot wheel in comparison with the head wheel of a centrifugal discharge elevator, the less the influence of centrifugal force in the boot, the lower the buckets pick up their load, and the less the energy wasted in the boot.

Large foot pulleys are better than small ones in another respect; they support the buckets better when the latter pick up their load. Fig. 264 shows, to scale, a bucket with 6-inch projection on a 6-ply belt running on a 15- or a 27-inch pulley. Since the bolt fastening is not rigid, the bucket moves in the direction of the arrow when it meets some resistance from the material, thus exerting a pull on the bolts until the bucket finds a backing

against the belt at some point A. The larger the pulley, the less the bucket will move and the smaller will be the pull on the bucket bolts. It will also gap away from the belt less at B and bits of material will not be so likely to catch and stick there and injure the belt.

Material Catching between Belt and Foot Pulley.—More elevator belts are injured from this cause than from any other. The pieces of material which cause the trouble may have been spilled from the buckets on the vertical run or on the head wheel; or lumps may fall from the head chute on a rebound, or when the material backs up in the chute from a choke or when the bin it supplies is full. When the feed chute is too high with reference to the foot wheel, or when it is set for "fly-feed," this trouble is more

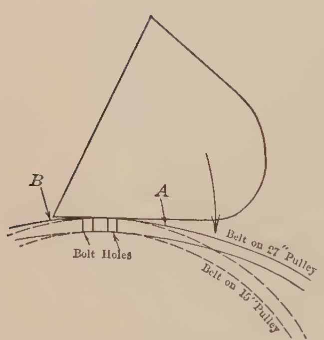


FIG. 264.—Better Support of Bucket on Foot Wheel of Large Diameter.

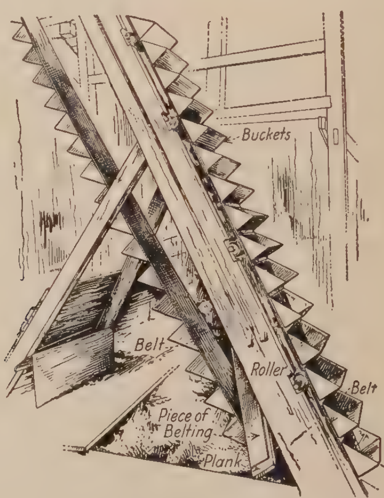


FIG. 265.—Deflector for Spill on Return Run of Elevator.

likely to occur than when the chute delivers to a sloped-front boot of proper design.

Guards placed close over the pulley do not cure the trouble altogether. If placed so close to the pulley as to shed falling pieces, they may serve to confine material which has gotten through the clearance space which must be left between the guard and the belt, and thus do more harm than good. If the elevator is inclined instead of vertical, spill from the head is less likely to fall on the foot pulley, especially if the casing is made, as in Fig. 257, with a vertical face on the descending side.

In some inclined elevators handling large pieces of stone, guards or shields have been used with success to deflect from the foot pulley any pieces of stone which may fall or roll down along the inside face of the descending belt. Fig. 265 shows such a device; it consists of a heavy plank set diagonally across the elevator above the foot pulley and provided

on its lower edge with several thicknesses of old belting which project far enough to keep close to the elevator belt and yet yield when the belt vibrates or sways, as it always does.

Devices like this need the intelligent interest of the men in active charge to keep them in working order. The plank may need adjusting from time to time, or the strips of belting may have to be replaced when worn. With such attention, a guard, like the one shown, may prevent serious injury to the belt and may pay for itself many times over in the longer life of the belt; without such attention, it may be discarded as a failure.

Elevators with no Foot Pulley.—The chances of injuring the belt would be fewer if belt elevators could be run without foot pulleys. This has been done in the works of a copper company in Arizona. Up to 1920, six or eight elevators had been put in at this plant, under the Cole patent of March 2, 1920 (Fig. 266), which covers the use of a belt elevator without a foot pulley but with a normally slack belt guided by a pulley on the rising side above the feed point. Some of these elevators had 24-inch belts about 10-ply with 16-inch buckets set staggered. "When running at the speed for a proper dumping effect for the buckets the lower end of the belt loop would take a form almost the same as that of the boot pulley and would take up the feed as well as if the usual boot pulley were used. The snub pulley would only come into play when an excess of feed was fed to the machine. There was no strain on the belt other than that of carrying the load; no sand or quartz particles were forced into the cover of the belt, the latter being clean all the time."[1]

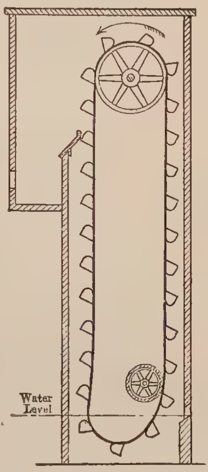


Fig. 266. — Crushed Ore Elevator with No Foot Pulley.

There are many elevators in which the load is so heavy that take-up tension must be added in order to get sufficient driving effect (see p. 299). In such cases, the foot pulley is a necessity.

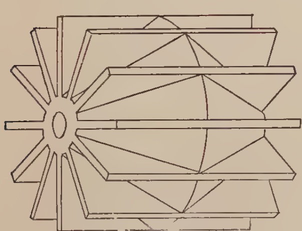


Fig. 267.—Foot Pulley with No Rim.

Special Foot Pulleys.—Several forms of pulley have been used to lessen the risk of injuring the belt. The Boss pulley shown in Fig. 267, presents less surface to the belt, and if the material is fine and dry it will be shed to each side of the belt by the conical surfaces in the pulley. Flanged pulleys in two sections have been used (Fig. 268); they offer less surface to the belt, but since the pulley has no crown, flanges are necessary to guide the belt. The flanges are bad because the edges of the belt tend to ride up on them and are worn, and the wheels are likely to act as a trap for material rather than a relief for the belt.

[1] Communicated to the author by Mr. David Cole.

Neither of these devices is of use when the material handled is in pieces that will wedge in the pulley. A piece of sharp stone wedged between the halves of the flanged pulley or between the arms or vanes of the Boss pulley may go around many times and hurt the belt in many places, while the same piece might go once around an ordinary pulley and then fall off.

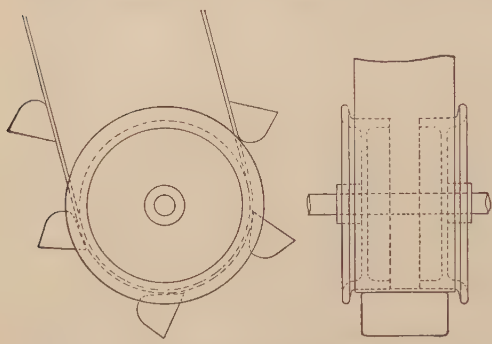


Fig. 268.—Double Flanged Foot Pulley.

It is possible that foot pulleys for thick and heavy belts on elevators handling ores and stone might be made with pneumatic or cushion rims that would be more yielding than the belt and still be rigid enough to guide the belt and apply tension to it.

The problem is a serious one, and one that is hard to solve. Men in the business have seen 10-ply belts cut clear through by stone jammed against the foot pulley; the force that will do this is hard to control by mechanical means. At present it is not attempted; the burden is put on the belt manufacturers, and they are making belts thicker and heavier, with outer covers and internal cushions of rubber to resist the blows and the pressure.

Ordinary Foot Pulleys for elevators handling coal and other materials which are not too abrasive are generally made in "double-belt" weight which for the usual range of diameters, from 15 to 30 inches and faces 6 to 18 inches, means that the edge of the rim is $\frac{1}{4}$ to $\frac{5}{16}$ inch thick at the edge and $\frac{3}{8}$ to $\frac{1}{2}$ inch thick in the center. The crown is usually standard—that is, $\frac{1}{8}$ inch on the diameter per foot of face. The face measures $\frac{3}{4}$ to 2 inches wider than the belt for the sizes used in elevator boots.

In handling sand and ores, the stir of material in the boot and the fine stuff adhering to the belt combine to wear out the rims of ordinary foot pulleys. For this service it is economical to order pulleys with rims thicker than standard, and with heavy arms, or with plate centers instead of arms.

Fastening Foot Pulleys to Shafts.—On this subject, see page 298.

Amount of Take-up Travel.—Take-ups for belt conveyors are often made with 3 feet or more of travel, so that 6 feet or more of belt stretch can be removed without cutting and resplicing the belt. Elevator belts are generally shorter than conveyor belts and the stretch is less, but aside from that, the range of travel of the foot shaft in a take-up boot must be limited between positions at which the buckets will pick up material properly and at which the feed chute will neither be choked nor yet swamp the foot wheel (see p. 287).

This circumstance limits the take-up travel in grain boots to 12 or 15 inches and in other belt elevators to 6 or 8 inches. In chain elevators the

take-up travel should be at least enough to permit the removal of one pitch of chain, if the links are all alike; of two pitches if the links are alternately inside and outside, as in some steel chains; and in continuous bucket elevators, at least enough to remove one bucket.

Lubrication of Boot Bearings.—Although oil is generally used for the bearings of grain boots (see p. 297), grease is preferable for most elevators handling coal, minerals, ores and rough materials. It is better suited to the cheap and simple bearings used in such machines, and in many cases it keeps dirt out of the bearings by forming a collar or ring where it squeezes out of the end. The best way to apply grease is to put the grease cup directly on the bearing (see Fig. 254 or Fig. 256); if the pit is deep or if one side of the boot is inaccessible, a pipe may be used to lead the grease to

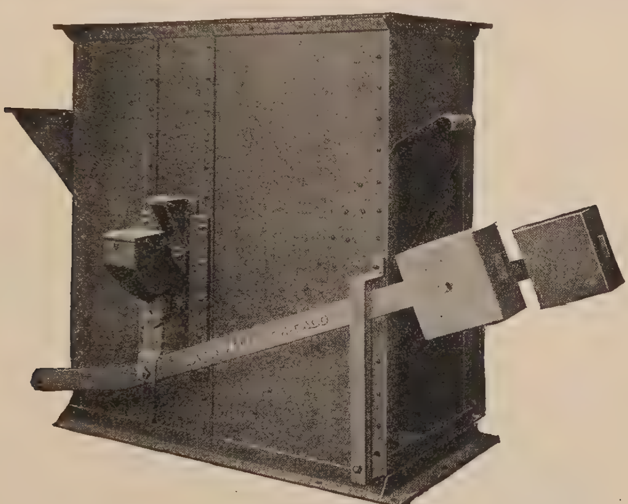


Fig. 269.—Automatic Take-up Boot with Grease Pockets on the Bearings.

the bearing. There is always some uncertainty about lubricating in this way; if the pipe is long, or is bent, or has elbows in it, the grease may clog in it in spite of screwing down the cap of the grease cup. Turning the cap may merely compress air in the pipe, or if the cap is small, it may turn so hard that with some classes of labor it will be neglected.

To make sure of lubrication in spite of occasional neglect, boots are made (Fig. 269) with large grease pockets cast on the bearings. The mass of grease rests directly on the shaft and feeds to it by gravity. Other boots have bearings made of hard wood impregnated with oil; these bearings require no lubricant, they are comparatively cheap and, when worn, they are thrown away. Fig. 270 shows an arrangement which has been used in the Western mining country. The boot shaft is a hollow brass shaft which contains a supply of oil; wicks feed oil to the bore of the foot pulley which runs loose on the shaft; the shaft itself does not revolve. In elevators

handling gritty ores, the pulley side of the belt may be worn off if the foot pulley does not revolve freely. The pulley in an ordinary boot may stick if the two take-up screws are not adjusted alike or if the two bearings bind the shaft because of dirt or lack of lubricant. When the pulley is loose on the shaft, as in Fig. 270, it is more certain to turn; there is less weight to be revolved and the turning of the pulley is not affected by the way the take-ups are adjusted.

In some elevators no attempt is made to lubricate the foot bearings. In hot-clinker elevators in cement mills the foot bearings are hard to get at, being in a deep pit where the heat is intense. It is good practice here to use chilled iron bearings and run them until they are worn out. It is more economical to do this than to use better bearings and then pay for oil, attendance and maintenance. A bearing which is not lubricated may

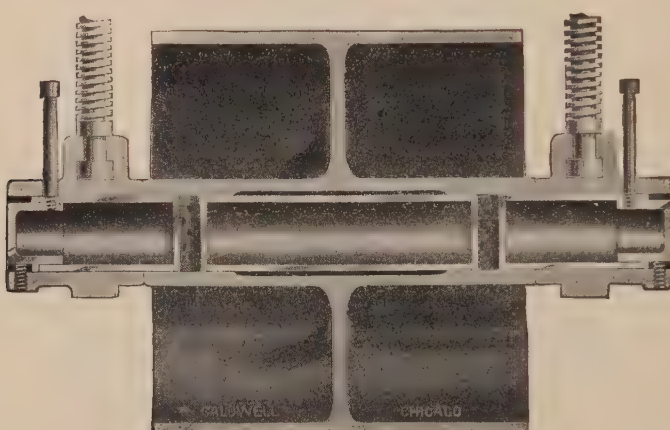


FIG. 270.—Loose Foot Pulley on Hollow Shaft with Internal Lubrication.

screech and get hot, but these faults are not serious at the foot of a clinker elevator. These elevators are chain machines, but the same principle also applies to some belt elevators. On this point, see page 285.

Grain Elevator Boots.—Old-time boots were of wood, made either as a prolongation of a wooden casing or pair of legs, or as a separate box on which the casing was mounted. The latter construction was more expensive, but the bearings were often mounted in a better way, the interior was more accessible and the boot was more easily cleaned. Wood boots are still used in some places; but they are subject to decay in damp pits, they get out of shape under the weight of high casings, and there is always the risk of a fire starting in a wood boot from a combination of oil-soaked wood and the heat from a bearing which has not been lubricated properly.

Modern grain boots consist either of the base portion of a steel casing or of a structure of cast-iron and steel plate on which a steel or wood casing is mounted. The former construction is often used in modern elevators

where the legs are very high, and where, on account of a large head wheel, the lower sections of the down leg approach the foot at a decided slant. Fig. 271 shows such a steel-plate boot with a 30-inch diameter foot pulley for a 40-inch belt. It is built as part of the casing with $\frac{3}{16}$-inch plate, 2-inch angles and $\frac{3}{8}$-inch bolts and rivets 4-inch pitch. Two $\frac{3}{16}$-inch removable slide plates give access to the inside of the boot, front and back. The shaft is $2\frac{7}{16}$ inches with 15-inch travel. The elevator is 196 feet center to center. The boot bearings slide up and down in a cast-iron frame bolted to the boot sides, and carry a slide plate which closes the slot on each side of the boot. A weighted frame mounted above the shaft provides an automatic take-up for slack and keeps a tension on the belt.

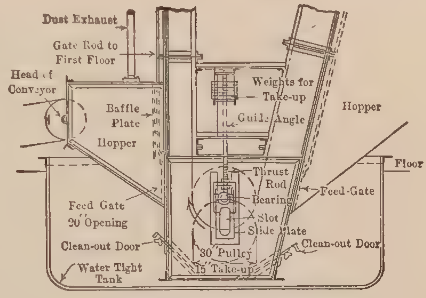


Fig. 271.—Foot of 40-inch Receiving Leg. Public Grain Elevator, New Orleans. (Ford, Bacon & Davis, Engrs.)

Cast-iron Boots.—Various styles of grain boots are listed by manufacturers. Generally they have cast-iron sides joined by cast-iron or steel plates curved to form the bottom of the boot. In the best designs, portions of the bottom plates are removable (Fig. 272) to give access to the

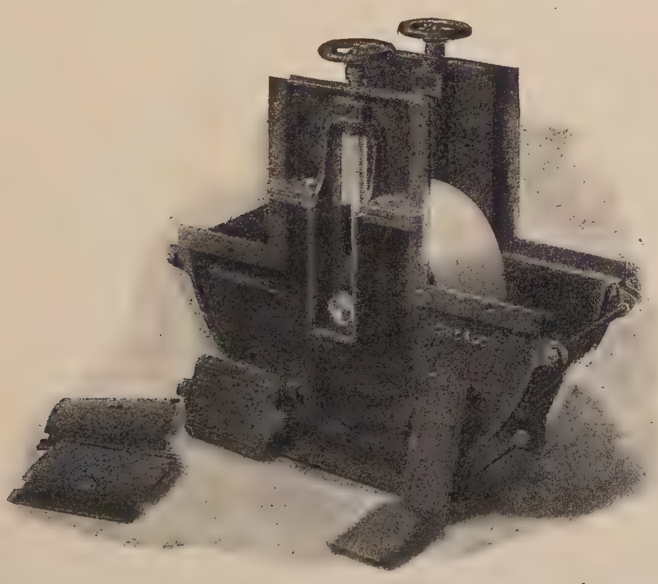


Fig. 272.—Cast-iron Grain Boot with Removable Bottom Plates. (Used with Wooden Legs.)

inside for cleaning or removing obstructions; sometimes only small clean-out doors in the side plates are provided, but these are too small to do much good when the boot chokes. At such a time, the grain is piled higher than the shaft and it is better to get at the trouble quickly by removing a section of the bottom so that a shovel can be used, and, if necessary, damaged buckets can be taken off the belt.

Fig. 273 shows a heavy cast-iron boot for a large grain elevator with a 24-inch pulley and 11 inches of adjustment. The bearings (Fig. 274) are closed at one end, and the opening at the other for the $2\frac{7}{16}$-inch shaft is sealed against the entrance of dust and dirt and against the leakage of oil by a felt washer and a packing ring. It carries a supply of oil in the base of the casting, enough for several weeks' run. A loose brass oil-ring delivers

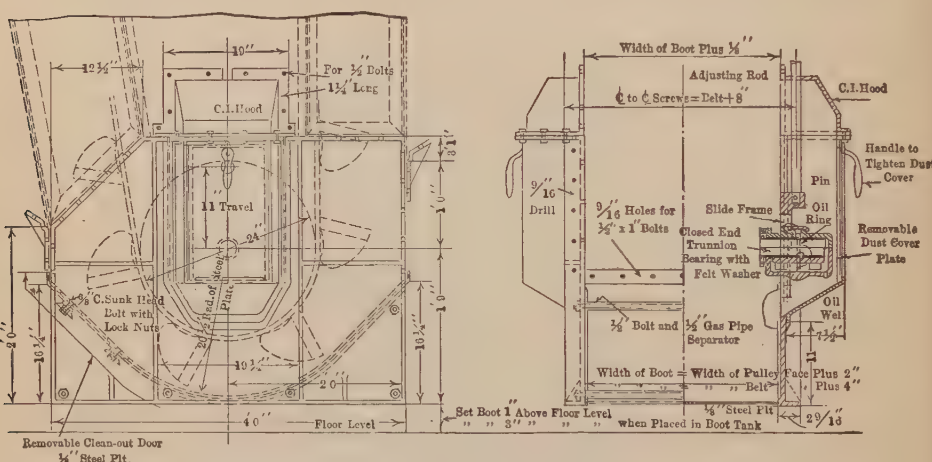


Fig. 273.—Heavy Cast-iron Grain Boot used with Steel Casing. (Witherspoon-Englar Co.)

oil to the shaft and a hinged cover permits the attendant to see whether the ring turns and the oil reaches the shaft. The trunnions or pivots of the bearing are carried in a slide casting, so that the shaft can be leveled and the belt made to run straight on the pulley independent of the setting of the boot. The slide casting is pinned to the adjusting screw and travels with it, the screw being moved by a handwheel which acts as a nut. The screw does not turn.

Lubrication of Grain Boots.—In cheaper boots the adjusting screw turns, is necked at the lower end and has a loose connection with the bearing itself, and in some of them the oil is fed to the bearing through the screw itself, which is then made of heavy pipe or hydraulic tubing. Experience with this method of oiling is not altogether satisfactory; dirt gets into the loose connection at the lower end of the screw and plugs the oil hole shut. Boots are often placed in deep and narrow pits and it may be desirable

to oil the bearings from the floor level, but in such cases it is better to use separate oil pipes screwed tight into the bearings and not oil through the take-up screws.

Oil is generally used for the bearings of grain boots, on account of the high speed of the shaft, often 100 r.p.m., and for cleanliness. The objection to grease lubrication is that the dirty collar of grease which accumulates at the ends of the bearings may fall off into the grain and spoil it for flour-milling.

In some ways it is better that the attendant should get down into the pit to oil the bearings; he can then see and feel the bearings and make an inspection of the boot. It is a common experience that boot bearings are apt to be neglected; then if the shaft should seize tight in a hot bearing,

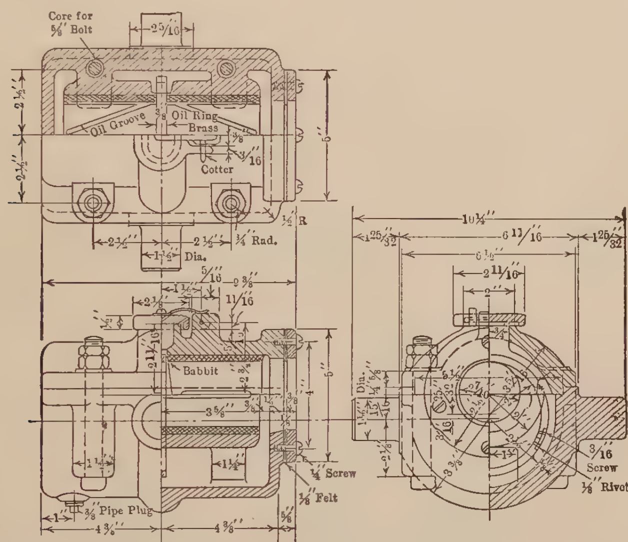


FIG. 274.—Self-oiling Trunnion Bearing for High-speed Grain Elevator Boot Shown in Fig. 273.

there is danger that the pulley may stand still and rub off the inner plies of the belt, or if the pulley is fastened only by set screws, it may turn while the set screws gouge grooves in the stationary shaft, or it may shift end ways on the shaft until its rim cuts through the side of the boot. These mishaps sometimes cause expensive shut-downs and then costly repairs. In the design of important grain elevators it is well to provide for good lubrication by using well-made self-oiling bearings, to make the pit large enough for access to both sides and both ends of the boot, and to keep the bearings outside the boot where they can be seen and felt. In the large elevator shown in Fig. 271 the foot shaft with its pulley, slides and automatic tension device weighed over 2200 pounds; a load of about 1100 pounds on each shaft bearing at about 100 r.p.m. represents a duty important enough to

deserve good bearings, thorough lubrication and regular inspection. As a precaution against heating, the foot bearings of elevators, as well as other bearings, are in some cases fitted with fusible metal plugs which make electrical contact and sound an alarm when the temperature of the bearing goes over 165° F. or some other established temperature.

Pulleys in Grain Boots are always made with a crown face and of the weight known in the trade as "double-belt." In small elevators it is sufficient to hold the pulley in place on the shaft by two set screws, but it is always an advantage, and in important work it is very desirable, to hold the pulley more securely. It is a common complaint that, in conveying and elevating machinery, set-screwed wheels will not stay in place, but shift on the shaft. The reason is that commercial shafting varies in diameter from its nominal size; it is seldom oversize but often .003 or .004 inch undersize. Pulleys in the trade are bored and reamed a few thousandths oversize to make sure of their going on commercial shafting; the result is that when a set-screwed pulley with an oversize bore goes on an undersize shaft with perhaps .005 to .01 inch clearance, it is very apt to shift, especially if the face is wide and the hub relatively short. Some men in charge of grain elevators (and other elevators) always "spot" the shaft, that is, drill shallow recesses in it to receive the points of the set screws; some use long set screws with lock nuts to keep them tight in the pulley hub; some keep the pulley in place by fastening collars with set screws to the shaft on each side of the pulley hub. A better way for pulleys and shafts of commercial grade is to key the pulley to the shaft with a fitted key and put two set screws over the key. A still better way, although it is seldom used for foot shafts, is to turn the shaft to an exact diameter, bore the wheel a few thousandths small and then press or drive it on.

Pulleys wider than 20-inch face should have double arms.

Pulleys for large elevators are generally made with closed ends, usually by fastening on steel-plate disks, so that the grain is kept out of the pulley and the arms do not add to the work of the elevator belt by stirring up the grain.

Automatic Boot Take-ups may be used (1) as a convenience to avoid attention to take-up screws; (2) to maintain driving contact on the foot pulley or sprocket wheel in those exceptional cases where an elevator is driven at the foot or where power is taken from the foot shaft; (3) in chain elevators, to prevent loose chain from climbing up on the teeth of the foot wheel; (4) in belt elevators, to maintain a tension T_2 in the down belt by applying a load to it which will act regardless of the normal stretch in the operation of the belt.

The last item is the most important, and it is the one to be considered here. For a discussion of it, see page 270. It is evident that if the load is light and the driving conditions at the head of the elevator are favorable there is little or no need for artificial tension in the belt; but if the load is heavy and if the belt is dirty, wet, or works in an atmosphere of dust, the coefficient of belt contact falls off and then there is need of artificial tension

to maintain the proper ratio of $\frac{T_1}{T_2}$. There is also need for artificial tension if the belt is heavily loaded even though clean; that condition, or an overload in the boot, has the effect of increasing T_1, and unless T_2 is great enough the belt will slow down and slip.

The great practical advantage of a weighted take-up on a belt elevator is that a steady artificial tension can be applied to the down belt which will keep T_2 high enough to prevent slip and avoid a choke in the elevator. On the effect of belt slip, see page 277.

Artificial Tension in Vertical Elevator Belts.—Table 56 has been prepared to show what artificial tension is necessary in vertical elevators for various conditions of drive and for various values of f, the coefficient of belt contact. For instance, an elevator handling a clean, sized material has a belt that weighs 2.3 pounds per foot; empty buckets weigh 4.3 pounds per foot and the material in the buckets 3.5 pounds per foot. Then the ratio $\frac{T_1}{T_2}=\frac{2.3+4.3+3.5}{2.3+4.3}=1.6$ and from Table 56 it is evident that no belt tension need be applied at the boot for any kind of drive. It is not

TABLE 56.—ARTIFICIAL TENSION FOR VERTICAL BELT ELEVATORS

T_1 = weight of loaded belt and buckets, plus pull to dig.
T_2 = weight of empty belt and buckets.

Tension to be applied at boot to maintain driving contact at head-pulley. Figures are percentages of T_2 to be added to T_1 and to T_2.

1	2	3	4	5	6	7
	Rubber-Covered Pulleys			Plain Iron Pulleys		
Ratios of $\frac{T_1}{T_2}$	$\frac{T_1}{T_2}=3.00$. Clean. $f=.35$	$\frac{T_1}{T_2}=2.33$. Dusty. $f=.27$	$\frac{T_1}{T_2}=1.87$. Wet. $f=.20$	$\frac{T_1}{T_2}=2.19$. Clean. $f=.25$	$\frac{T_1}{T_2}=1.87$. Dusty. $f=.20$	$\frac{T_1}{T_2}=1.87$. Wet. $f=.20$
1.8	0	0	0	0	0	0
1.9	0	0	3	0	3	3
2.0	0	0	15	0	15	15
2.1	0	0	26	0	26	26
2.2	0	0	37	1	37	37
2.3	0	0	49	9	49	49
2.4	0	5	60	17	60	60
2.5	0	13	72	26	72	72
2.6	0	20	84	34	84	84
2.7	0	28	95	43	95	95
2.8	0	35	107	51	107	107
2.9	0	43	118	60	118	118
3.0	0	50	130	68	130	130
3.1	5	58	141	77	141	141
3.2	10	65	153	85	153	153
3.3	15	73	164	93	164	164
3.4	20	80	175	102	175	175
3.5	25	88	187	110	187	187

necessary to lag the head pulley, and an iron pulley might be expected to drive the belt even though the surfaces in contact were wet, provided the work of picking up material from the boot were not great. If the buckets carried a heavier material, or if the pick-up were harder, the ratio $\frac{T_1}{T_2}$ might rise to 2.0 and then it would be necessary, for wet conditions, to load the down belt (and necessarily the up belt) at the boot with about 15 per cent of the dead weight of the down run. That is, if the empty belt and buckets on the down side weighed 400 pounds, then the added load at the boot (considering both runs of belt) would be $400 \times .15 \times 2 = 120$ pounds.

When the buckets must dig their load from the boot the pull in the up belt is more than the weight of the belt and loaded buckets by the pull required to dig. In grain elevators the buckets, and often the belt, are relatively light; this factor, on the one hand, and on the other the considerable pull required to dig, combine to raise the ratio of $\frac{T_1}{T_2}$ to 2.5 or more. In the example quoted in Chapter XX the ratio was 2.63, and from Table 56 it is evident that under the operating conditions—a rubber-covered head pulley working in a dust—it would be necessary to apply to the belt in the boot a load of 22.5 per cent $\times 2 = 45$ per cent of the weight of the down belt with its buckets, or $2700 \times .45 = 1215$ pounds.

Varying the Boot Tension with the Load.—When the operating capacity of a grain elevator is based upon not more than 80 per cent of the level-full capacity of the buckets, as rated in the catalogues of manufacturers, there is generally margin enough to provide for ordinary contingencies of feed; but if the feed is irregular and subject to fluctuations beyond the normal rate of loading, or if the capacity of the elevator is to be forced at times beyond 80 per cent of the rated capacity of the buckets, then provision should be made to apply extra belt tension at the boot, if necessary, so that the ratio of $\frac{T_1}{T_2}$ does not become too large for the driving conditions at the head of the elevator.

Illustration.—In the elevator, the boot of which is shown in Fig. 271, the empty belt and buckets on the down leg weighed about 4200 pounds and for a capacity of 20,000 bushels per hour the loaded belt on the up leg weighed about 10,000 pounds; allowing 800 pounds for the digging and friction in the boot, the ratio of belt tensions is $\frac{10,800}{4200} = 2.57$, and from interpolation in Table 56, Column 3, the total weight to be applied to the boot is $4200 \times .179 \times 2 = 1500$ pounds. For a capacity of 22,500 bushels per hour the ratio is $\frac{11,500}{4200} = 2.74$ and the weight required is $4200 \times .308 \times 2 = 2600$ pounds. For the maximum rating of the elevator, 25,000 bushels per hour, the ratio is $\frac{12,200}{4200} = 2.9$ and the weight required is $4200 \times .43 \times 2 =$

3600 pounds. These values are, of course, based on the assumed coefficients of belt contact as stated in Table 56, but they agree quite well with the best practice in grain-elevator design and operation.

On the permissible unit stresses in elevator belts, see page 274.

Varieties of Automatic Boot Take-ups.—A simple way is to make the foot pulley heavy enough, or add weights to the boot shaft, to take up the slack or maintain the proper tension on the belt. In some belt elevators with 36-inch 10-ply belts, used by a Western copper-mining company, the foot pulleys are 42 inches in diameter with rims $2\frac{1}{2}$ inches thick. The foot bearings slide in guides, and no take-up screws are used. An old device is a pair of levers, one on each side of the boot, pivoted at one end, weighted at the other end and with an intermediate link bearing down on top of the boot bearing. One arrangement is shown in Fig. 275. Several patents of no worth have been issued on arrangements where the whole boot is hanging on the loop of belt and is guided by sleeves or tracks on the lower part of the elevator casing. A few elevators have been built where the whole weight of the boot is carried on a pivoted arm so as to maintain a tension on the elevator belt or chain, and since the wheel does not travel within the boot, a fixed distance is preserved between the sweep of the buckets and the bottom of the boot. This is an advantage in handling lumpy materials and materials like fertilizers, which form a hard, dense crust beyond the sweep of the buckets, but it is of no value in handling grain or similar substances (see Fig. 258).

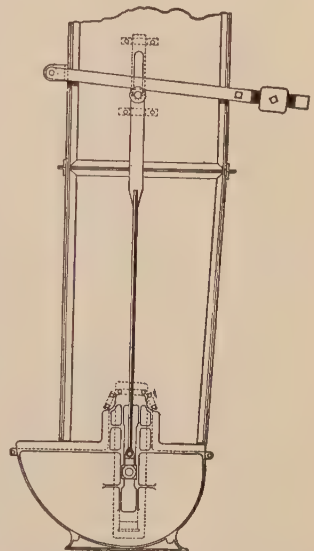


FIG. 275.—Weighted Lever Device for Automatic Take-up.

Edmond Take-up.—A practical device used on many large grain elevators is the Edmond automatic take-up (Edmond-Norell patent of 1911). It consists of a weighted frame, like that shown in Fig. 271, for maintaining the tension on the belt, plus means for independently moving the foot bearings in the frame so as to adjust the level of the shaft and cause the belt to run true on the foot pulley. In practice, the lead of the down belt on to the foot pulley is also controlled by adjusting the deflector pulley which in large elevators is placed in the down leg where it leaves the vertical to slant to the boot. A very slight deviation of this pulley from true level will train the belt one way or the other on the foot pulley.

Fig. 269 shows a belt elevator boot in which the belt tension is maintained by a weighted lever device.

Chokes in Grain-elevator Boots are due to various causes:

1. Elevating capacity of the buckets is too small for the feed.

2. Bins become full; grain backs up in head chute, then falls down back leg.

3. The supply of current to the motor is interrupted and the elevator stops suddenly; or the voltage falls, and the motor fails to pull the load; or there is a slip or failure in the power transmission.

4. Sticks, tools, etc., get into the boot, foul the belt and prevent it from moving.

5. Upper end of discharge chute is set too high for a clean discharge under all conditions.

6. Strings or pieces of bagging catch on the upper end of the chute, prevent a clean discharge and cause spill down the back leg.

7. If the elevator is shut down while loaded and is not fitted with a back-stop, the elevator may run backward and fill up the boot and lower part of the casing.

8. Speed is too high, grain hits the top or front of the hood and falls down the back leg.

9. The elevator is stopped before the supply of grain to the boot is shut off.

10. Take-up tension is not maintained. (On this point, see p. 270.)

When the choke is such that the belt slows down or stops while the power is still on and the head pulley continues to turn, the result is that the pulley rubs hard on the inside of the belt, wears the plies away and often generates heat enough to start a fire and cause a dust explosion. A report on the cause of a disastrous explosion says:

"There was no question after investigation but that a choke had occurred in an elevator leg at ten minutes to twelve. The men left the plant shortly after, and on their return at noon smelled the odor of burning rubber, which they thought was due to a hot belt in the basement, a belt which had been rubbing to one side. We found a man who had been to the top floor of the elevator and only about two minutes before the explosion saw smoke coming out of the elevator, and saw the flames of the burning belt, and only had time to get to the first floor before the explosion. . . . There was no general fire; when men entered the plant less than half an hour after the explosion, that particular elevator leg was red from fire; it was the only one in which there was fire. The belt was burned in two." [1]

In many chokes the belt slips but does not stop; in this case the pulley side may be seriously torn and frayed by the continued rotation of the head pulley, especially when the lagging or covering of the pulley is in bad condition, with bolts projecting (see p. 250). The heating of the belt due to frequent slipping of this kind may, in combination with excessive creep due to high belt tension (see p. 275), tend to "age" the friction in rubber belts or cause canvas belts to dry out and crack. This is more true of oil-

[1] Dr. H. H. Brown, U. S. Dept. of Agriculture, reporting to the U. S. Grain Corporation on the cause of an explosion in a grain elevator at Port Colborne, Ontario, in which ten men were killed. (Proceedings of Conference on Grain Dust Explosions, April 24, 1920.)

saturated belts (Class 1 impregnation) than of belts saturated with asphaltic compounds; the latter resist heat better and are not affected to the same degree.

Foot Pulleys Too Small.—It is certain that much of the prevalent trouble with choked grain boots is due to the use of foot pulleys that are too small. When a boot pulley revolves three or four times as fast as the head pulley the forces which prevent the grain from staying in the buckets under the foot wheel are five to seven times as great as those which throw the grain out of the buckets on the head wheel (see p. 214). This prevents the buckets from taking any load until they are on the point of leaving the foot wheel or are on the straight lift. All the while they are in contact with the foot pulley they merely stir up the grain, add to the pull on belt and bolts and consume power. When the pulley is larger, the conditions for pick-up are better (compare Figs. 202 and 204), the boot is larger and there is more room in it to take care of an accidental accumulation of grain; the excess of feed over elevator capacity, or what spills down the back leg, will not pile so deep on the rising side; the work of digging will be easier and chokes will not be so frequent nor so harmful.

Lack of Automatic Take-up.—For reasons stated on pages 266 and 270, grain-elevator belts will slip if the load is heavy and if the belt tension is not sufficient. As between no load and full load, a belt may stretch some inches in 100 feet; if an automatic take-up is not used, and if screws are used to tighten the belt, neglect to adjust the screws during the operation of the elevator may cause the ratio of belt tensions $\frac{T_1}{T_2}$ to fall to such a point that the head pulley will not drive the belt. Then the belt slips and a choke occurs. The same thing may happen if the weight on an automatic take-up is not sufficient; but in any case, an automatic take-up is better than a screw take-up on any grain elevator; in high elevators it is really essential.

Prevention of Chokes in Grain Elevators.—In the design of the elevator the bucket capacity should be larger than is necessary to give the required number of bushels per hour; for reasons stated on page 235 it should be great enough to cover the peak-load capacity which, for a minute or a few minutes, may be at a rate sufficient to exceed the hourly rate by a large margin. An excess of bucket capacity is a kind of insurance; it takes care of emergencies, and if a choke does occur, the elevator can dig itself clear in a short time without damage to the belt or buckets.

The lip of the head chute should come close to the buckets (Fig. 290) and it should be set not less than 15° below the center of the head shaft if the belt travels at the speeds of Table 35; 20° is better. If the belt speed is lower, Table 36, the angle should be 30° for grain. Many high-speed elevators have chutes set 12° or 15° below the head shaft; but there is always the risk that the speed may fall off, or that, for some other reason, the buckets may not discharge properly into a chute set too high. To set the lip of the chute at 20° instead of 12° means an increase of only 7

or 8 inches in the height of an elevator with a 72-inch head pulley—not a heavy price to pay for some insurance against trouble.

There should be plenty of room in the front of the hood over the head pulley, so that the direct discharge and also the splash from the buckets will enter the chute without striking the front plate (see Fig. 290).

There are in use a number of safety devices to prevent grain from dropping down the back leg in case the flow from the discharge spout is stopped by the chute filling up or by a choke in the spout itself. An auxiliary by-pass spout is fitted to the discharge spout near the head casing so that when the main spout fills up, the grain will enter the by-pass and be directed to an emergency bin, or be dropped on a floor where it can be seen, or made to enter a counterbalanced tank, which, on receiving a certain weight of grain, drops, sounds an alarm, and closes the gate at the boot or else stops the conveyor which delivers to the boot.

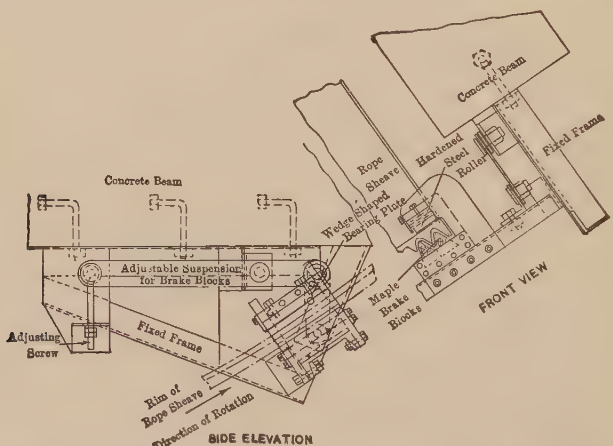


Fig. 276.—Reverse-motion Brake for Elevator Head Sheave. (Witherspoon-Englar Co.).

On the influence of automatic take-ups and large foot pulleys in reducing the risk of chokes in the boot, see above.

When a choke does occur it is important that the rotation of the head pulley should stop as soon as the belt slows down; otherwise the belt may be damaged or burnt. When the drive is from a separate electric motor the danger of a burnt belt is reduced when the circuit breaker has an overload release set to throw out before the overload becomes too great. Electrical safety devices are used also to stop motors when bins are full, to stop the conveyor which delivers to the boot as soon as current is cut off from the elevator motor, and to prevent the conveyor motor from starting until the elevator is up to full speed.

Elevator Back Stops.—To prevent the elevator from running backward under load with the power shut off there may be on the end of the head shaft a ratchet wheel which engages a pawl fulcrumed on the beam that supports

one of the shaft bearings. To avoid noise, the pawl is usually fitted with a friction device of some kind which keeps it away from the ratchet teeth while the shaft revolves in the operating direction, but allows it to drop into place and stop the ratchet wheel if the shaft should start to turn backward.

Fig. 276 shows a back stop applied to a rope sheave on an elevator head shaft. A hardened steel roller is contained in a frame on which is mounted a brake shoe, engaging one or two of the grooves on the lower or empty side of the sheave. The roller bears against a finished part on the inside of the sheave rim and is backed up by a wedge-shaped steel plate. In normal rotation, the roller stays at the small end of the wedge plate, and the brake block hangs clear, but if the sheave starts to reverse, the roller travels toward the thick end of the wedge plate and brings the brake block into contact with the grooves of the sheave.

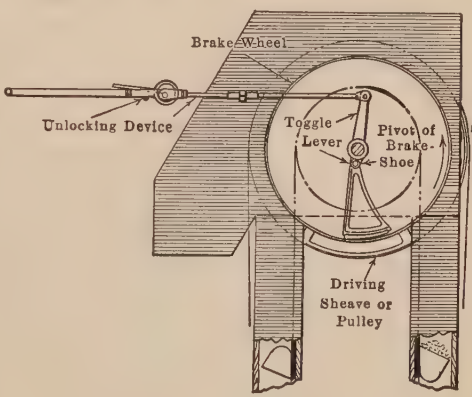


FIG. 277.—Gemlo Back-stop.

Fig. 277 shows the Gemlo back-stop (patented 1916). On the inside of the rim of the brake-wheel which is keyed to the head shaft there rests a brake shoe. The shoe is jointed to a two-armed lever which is loosely mounted on the head shaft. During normal rotation the shoe hangs loose, but if the shaft starts to reverse the shoe takes hold and forms a toggle-lock with the two-armed lever to prevent rotation. To unlock the brake, the long arm of the two-armed lever is moved so as to break the toggle-lock.

CHAPTER XXII

INCLINED ELEVATORS

Inclined Elevators.—For the same belt speed and same size of foot wheel the pick-up at the foot of an inclined centrifugal discharge elevator is apt to be better than in a vertical elevator, because at the moment of leaving the wheel to enter on the straight inclined run, the bucket is tilted back and the resultant of pressure due to the combination of centrifugal force and gravity is directed more toward the bottom of the bucket and is not so likely to force material out over the front lip. This is not so noticeable when the mass of material in the boot is small and when the buckets do not fill; but when the mass is deeper, and the loading continues until the lip of the bucket is at or above the level of the foot shaft, then the last material delivered to the bucket is jerked toward the back when the bucket enters on the straight inclined run and does not partly spill over the front lip, as happens when the run is vertical.

Fig. 278 illustrates this difference by showing that for speeds and sizes of wheels as shown in Fig. 206, the resultant pressure for the bucket on a 20° incline makes an angle of 32° with the back of the bucket, while the pressure for a bucket on a vertical run is 15 per cent greater in amount and inclined at 44° to the back.

After passing the foot wheel, the belt and buckets vibrate to some degree, and the material, if free-flowing, tends to shake down to a surface

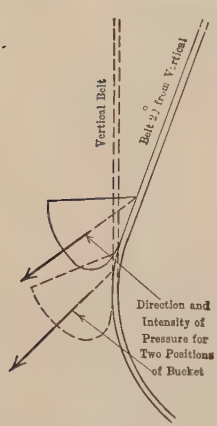


Fig. 278.—Conditions for Filling Buckets at Foot Wheel more Favorable when Elevator is Inclined.

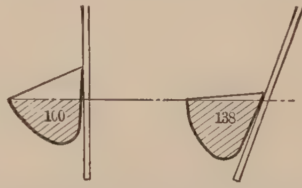


Fig. 279.—Comparison of Loading Malleable Iron "A" Buckets on Vertical Run or 20° Incline.

at right angles to the direction of gravity—that is, the surface of the material in the bucket will be approximately horizontal. This means that a bucket piled full on leaving the foot wheel is apt to spill less on an inclined run

than on a vertical run. Fig. 279 shows that the water-level capacity of a standard malleable-iron Style A bucket is 38 per cent more on a 20° incline than on a vertical run. While this is not true to the same extent of materials which do not flow freely and which pile up at steep angles, still it is generally true that standard Style A buckets carry better on inclines up to 30° from the vertical than on the vertical.

When the bucket reaches the head wheel the material in it comes under the action of centrifugal force, and if the speed is high enough, and if the material lies up to the lip of the bucket, some of it may spill over the lip. Whether this happens or not depends on the shape of the bucket and the direction of the resultant pressure at the point where the belt meets the wheel. Reference to Figs. 202 and 205 will show that the pressure at position 4 in ordinary centrifugal discharge elevators is less likely to cause

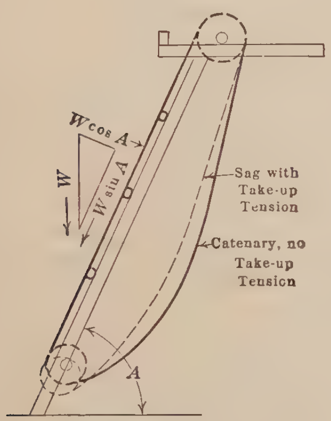


Fig. 280.—Hang of Return Belt on Inclined Elevator as Affected by Take-up Tension.

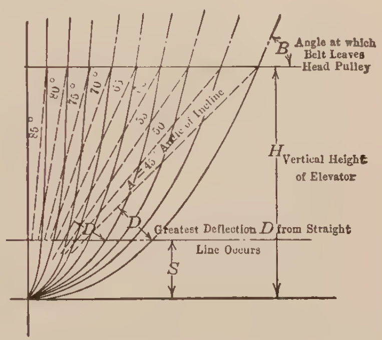


Fig. 281.—Forms of Catenary Curves for Return Belt for Different Angles of Slope.

spill than the pressure at position 3, because it is less in intensity and is directed more toward the bottom of the bucket and not so much toward the lip. The same thing is true of any position on the wheel between 3 and 4 where an inclined belt meets it; hence in an inclined elevator a bucket loaded to the edge of the lip is less likely to spill on reaching the head wheel than the same bucket in a vertical elevator with the same size head wheel and same belt speed.

Path of Belt on Inclined Elevators.—Fig. 280 represents an inclined belt elevator with the up run supported on idlers and the down run hanging free. If no tension is applied to the belt at the foot wheel, the down run forms a catenary. Fig. 281 shows the forms of catenary curves for elevators inclined from 45° to 85° from the horizontal, and Table 57 shows characteristics of these curves. The use of the table can be understood from an example taken from practice. In a heavy stone elevator inclined 65°

TABLE 57.—FACTORS FOR INCLINED BELT ELEVATORS WITH UP RUN SUPPORTED AND RETURN RUN HANGING FREE (NO ADDED TAKE-UP TENSION)

1	2	3	4	5	6	7	8	9	10
Angle A	$L=$ Factor for Length of Down Belt	$K=$ Factor for Tension in Down Belt	$S=$ Factor for Locating Maximum Sag	$D=$ Factor for Deflection of Down Belt	Angle B	Angle of Belt Wrap	Ratio $\frac{T_1}{T_2}$ for $f=.25$	Ratio $\frac{T_1}{T_2}$ for $f=.35$	Sin $A+$ f cos A $=$Factor for W
45	1.50H	1.62wH	.26H	.20H	67° 34′	157° 26′	1.99	2.62	.78W
50	1.39H	1.46wH	.26H	.19H	71° 36′	158° 24′	2.00	2.64	.82W
55	1.30H	1.34wH	.26H	.18H	75° 9′	159° 51′	2.01	2.66	.85W
60	1.23H	1.25wH	.25H	.16H	78° 23′	161° 37′	2.03	2.69	.89W
65	1.17H	1.18wH	.25H	.14H	81° 7′	163° 53′	2.05	2.71	.94W
70	1.12H	1.13wH	.24H	.12H	83° 32′	166° 29′	2.07	2.77	.96W
75	1.08H	1.08wH	.23H	.10H	85° 39′	170° 29′	2.10	2.83	.98W
80	1.05H	1.05wH	.22H	.07H	87° 28′	172° 33′	2.13	2.87	.99W
85	1.02H	1.02wH	.20H	.04H	88° 57′	176° 03′	2.16	2.93	1.00W
90	1.00H	1.00wH	....	....	90°	180°	2.19	3.00	1.00W

w = weight of empty belt and buckets per foot (pounds).
W = total weight of belt buckets and material on loaded run (pounds).
H = vertical height in feet.

from the horizontal, the belt was 38-inch 10-ply and weighed 12 pounds per foot; steel buckets empty weighed 50 pounds per foot; the height on the incline was 67 feet and the vertical height 60 feet.

From Column 2, the length of belt from where it leaves the head pulley to where it meets the foot pulley is $1.17 \times 60 = 70.2$ feet.

From Column 3, the pull in the return belt at the top is $1.18 \times 60 \times (50+12) = 4390$ pounds.

From Column 4, the greatest sag occurs at a point in the belt $.25 \times 60 = 15$ feet vertically above the bottom of the foot pulley.

From Column 5, the deflection at that point from a straight line touching the two pulleys is $.14 \times 60 = 8.4$ feet (measured at 90° to the straight line).

Column 6 shows that the return belt is inclined at 81° from the horizontal (9° from vertical) where it leaves the head pulley and from this we find that the angle of belt wrap on the head pulley is about 164° (see Column 7).

When by means of take-ups or adjustable bearings, tension is applied to the belt, the curve changes its shape, becoming more like a parabola; the deflection D becomes less and the point of maximum sag rises slightly. In practice it is generally necessary to put tension in the return belt to stop it from swaying too much and also to increase the driving effect at the head pulley by making T_2 larger (see p. 309). Nevertheless, the catenary should always be laid out to find the length of the belt, to locate the lip of the head chute and to show what clearance is required back of the elevator in case the belt should run slack.

On the assumption that the curve is a parabola, the deflection of the belt from the straight line can be obtained thus: In Fig. 282 divide the straight line into 4 equal parts. Calculate the pull in the return belt at the top pulley from Column 3, Table 57, add to it the tension applied to the return belt by the take-ups and call the sum T_2. Then if s = *horizontal* distance over which the belt hangs in feet and w = weight per foot of return belt and buckets, then the vertical deflection CD, in feet, at the middle point is $h=\frac{Ws^2}{8T_2}$ (see Kent's M. E. Pocket-Book). The vertical deflection at the quartering points G and F is $\frac{3}{4}h$.

To draw the curve, locate E at a distance h below D; then BE will be tangent to the upper part of the curve where the belt leaves the pulley and AE will be tangent to the lower part where it goes on to the foot pulley.

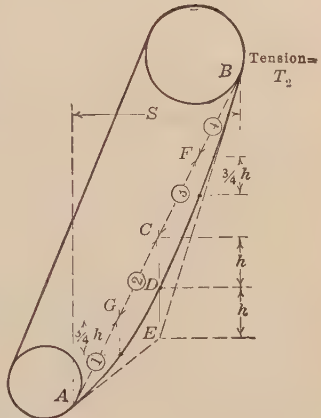


FIG. 282.—Lay-out of Return Belt as a Parabola.

Calculation of Applied Tension.—If in Fig. 280 W represents the total weight of the loaded up run, belt, buckets and material carried, then the downward component of W which puts a direct pull in the belt is W sine A, and the component which makes the idlers turn is W cosine A (see also Fig. 99); if f is a coefficient which includes the friction of the belt idlers and the slight lifting of the load in passing over the idlers, then fW cosine A is the belt pull due to contact with the idlers. The total belt pull due to the weight is, therefore,

$$\text{Pull} = W(\text{sine } A + f \text{ cosine } A).$$

To this should be added the pull due to pick-up (see p. 273).

Column 10, Table 57, gives values of (sine $A+f$ cosine A) in which f varies from .05 on steep elevators to .10 for an incline of 45°, the higher coefficients allowing for the greater motion of the belt and load in passing over idlers where the incline is not steep. The figures in Column 10 assume that the material is fed to the buckets from a chute and is not dug from a boot.

In large belt elevators handling stone and ore in continuous buckets the weight of material carried is great in proportion to the weight of the empty belt and buckets; and for reasons already mentioned in Chapter XX it is necessary to tighten the belt by means of the take-ups. In the stone elevator referred to above the buckets held at full capacity 130 pounds of stone per linear foot. The length of the inclined loaded run is 67 feet; its weight is therefore $67(12+50+130)=12{,}864$ pounds and from Column 10, Table 57, the pull at the head due to the weight is $.94\times 1{,}2864=12{,}093$ pounds. For the pick-up of material we may allow the equivalent of

11 feet on the lift (see p. 273) or 1430 pounds, making a total pull $T_1 =$ 13,523 pounds.

Since the pull at the top of the return belt is 4390 pounds, the ratio of $\frac{T_1}{T_2}$ for a natural sag of the return belt without applied tension is $\frac{13{,}523}{4390} = 3.08$. This is greater than 2.71, which is the working ratio of $\frac{T_1}{T_2}$ when $f = .35$ (see Column 9), hence not even a lagged pulley will drive the elevator unless some take-up tension is added. To find out what tension x is necessary, assume that the head pulley is lagged and that the ratio of $\frac{T_1}{T_2}$ is to be 2.7, then

$$13{,}523 + x = 2.7(4390 + x)$$

from which $x = 980$ pounds.

When the take-up puts 980 pounds tension into each run of the elevator the pull in the up belt is $13{,}523 + 980 = 14{,}503$ pounds and the 36-ounce duck is stressed to $\frac{14{,}503}{38 \times 10} = 38$ pounds per inch per ply.

Discharge from Inclined Elevators with Spaced Buckets.—In a vertical elevator some clearance space must be left between the upper edge of the discharge chute at the head and the lips of the descending buckets. Although the space may be small, some drip from the buckets, or material delayed in discharge, may fall through it and be wasted; but if the elevator is inclined, the chute may be placed partly under the path of bucket travel so that some or all of the spill may be caught.

This point is not of much importance in handling free-flowing materials like grain. These can be picked up and discharged at high speed (see Table 35); but if the material is damp and sluggish, or if it is hard, lumpy and abrasive, lower speeds like those of Table 36 are preferable; or it may be advisable to pick up and discharge difficult material at speeds still lower than Table 36. If the elevator is run at speeds lower than Table 36 it *must* be inclined in order to catch the spill and the delayed discharge. This condition of discharge is illustrated in Fig. 214 and it is shown also in Fig. 283. If Fig. 283 is turned so that the line A—B is vertical, the view represents an elevator inclined at 40° from the vertical and run at a speed less than Table 36. A chute at C will catch the material which begins to fall from the buckets high on the wheel, and also some of what is scattered by interference between bucket D and the discharge from bucket E.

Size of Head Wheels.—The bad discharge caused by the interference mentioned above causes a waste of material and power, because even in an inclined elevator a chute set partly under the head will not catch all of the scattered material. It is evident from Fig. 283 that if the head wheel were 18 inches in diameter instead of 36 inches the buckets which come every 12 inches would be 76° apart on the wheel instead of 38° apart as

shown in the figure. With this greater angle between the adjacent buckets, the discharge would be cleaner and interference would not occur.

This is shown better in Fig. 284, representing the head of an elevator inclined 30° from the vertical. With the large wheel, the positions of consecutive buckets are 1, 2, 3, etc. With the same linear spacing of buckets on a head wheel of half the size the corresponding positions are 1, 4, 5, etc. If the speed is such that discharge begins at 1, and follows the

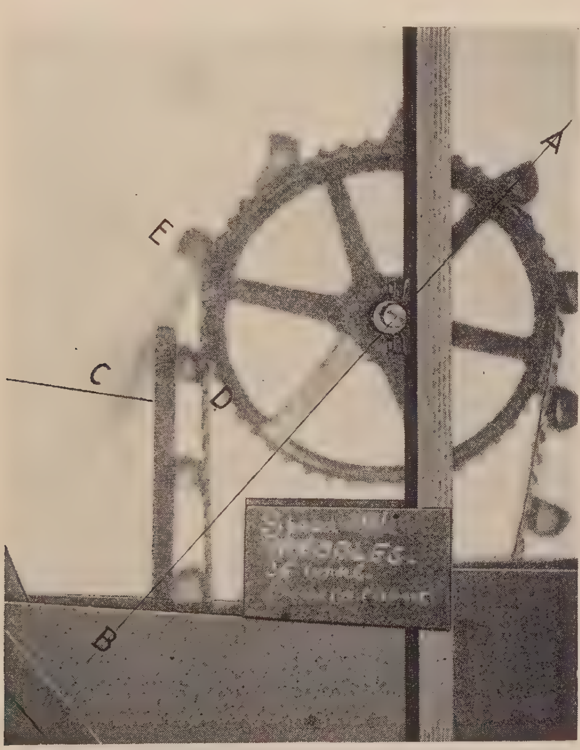


Fig. 283.—Condition of Discharge for which it is Necessary to Incline the Elevator.

parabola 1—G, some of it will hit the bucket at 2; but such a discharge would clear the bucket at 4 just leaving the small head wheel.

Discharge with Small Head Wheel.—As a matter of fact, the discharge from a bucket at 1 on the small wheel does not follow the path 1—G, but goes out on the parabola 1—P, because for a given belt speed, centrifugal force at a head wheel varies inversely as R—that is, it is twice as great on a wheel of half the size, *for a given belt speed.* The two arrows drawn from 1 represent by their position and their length, as drawn, the direction and intensity of the resultant forces which cause material to leave the bucket at 1, and the parabolas are tangent to them.

This explains why, in an inclined elevator, run at a given speed in feet per minute, a small head wheel makes a better discharge:

1. The buckets turn more quickly under the wheel and get out of the way of discharged material.

2. Centrifugal force is more active and there is more "throw" to the discharge.

Point at which Discharge Begins.—In considering the action at the head of an inclined elevator run at comparatively slow speed, the point at which discharge begins may be referred to. According to theory, the material begins to leave the bucket on the descending side of the wheel at the point at which centrifugal force equals the radial component of the weight. In Fig. 285 the forces which determine discharge are (1) gravity, acting downward with a force W, or if measured on the radial line,

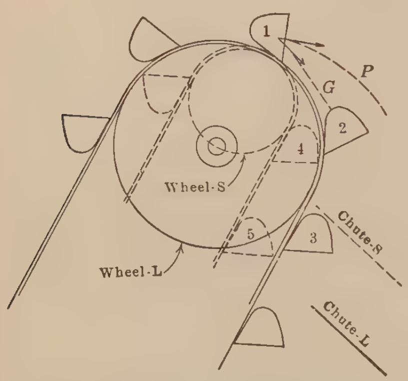


FIG. 284.—Effect of Wheel Diameter on Discharge at Head of Inclined Elevator.

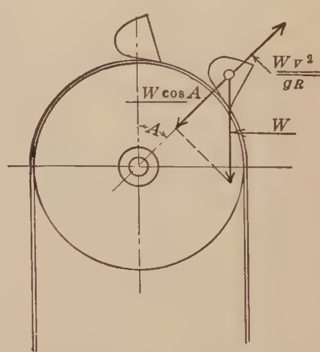


FIG. 285.—Point at Which Discharge Begins.

W cosine A, where A is the angle from the vertical; (2) centrifugal force, acting radially outward with a value $\dfrac{Wv^2}{gR}$ (see p. 212). The mass within the bucket will be in equilibrium, tending neither to fall out nor fly out when $\dfrac{Wv^2}{gR} = W$ cosine A or when cosine $A = \dfrac{v^2}{gR}$.

When $v^2 = gR$, as at the high speeds of Table 35 (see p. 212), cosine $A = 1$, $A = 0°$, and discharge is ready to begin at the top of the wheel; when $v^2 = \frac{2}{3}gR$, as at the lower speeds of Table 36 (see p. 216), cosine $A = \frac{2}{3}$, $A = 48°$, and discharge may be said to begin about halfway down in the quadrant of discharge. It is a fact, however, that some material always leaves the bucket sooner than would be indicated by the angle A calculated in this way. It is caused by the shifting of the load within the bucket; at position 4, Fig. 205, the resultant pressure is directed toward the bottom of the bucket; at 6 it is directed toward the back; since the surface of the material tends to arrange itself at right angles to the line of pressure, the

load shifts in the bucket; at 6 it flattens out on the back of the bucket, and if the bucket is full or nearly full some of the material may spill out over the leading edge of the back. This can be counteracted by making buckets with high backs or hooded backs, but ordinary buckets on belt run at slow speed are always apt to spill if they carry a full load.

Spill due to Shift of Load within the Bucket.—This spill is not so likely to occur in high-speed elevators, because discharge occurs before the load has time to shift in the bucket, and if it does occur, it may be affected by centrifugal force to a degree sufficient to describe a parabola that will land it in the head chute; but in elevators run at speeds less than Table 36, any material spilled between the top of the wheel and the point of theoretical discharge will fall so nearly vertical that the only way to catch it is to incline the elevator and set the chute well under the head wheel. The loss of some of this material is unavoidable, being due to interference with the descending buckets, but some material will enter the chute.

Size of Head Wheels and Spacing of Buckets.—When, for the sake of slow speed in pick-up or discharge, inclined elevators are run at speeds one-half of those in Table 35, the effect of centrifugal force becomes very small and the main discharge from a bucket occurs more than halfway down in the discharge quadrant—that is, angle A, Fig. 285, is over 70° from the vertical. The path of discharge is then nearly vertical, and in order to avoid interference the leading bucket should be so far ahead on the arc of travel that the discharged material cannot fall on it. The spacing of buckets in slow-speed inclined elevators can therefore best be expressed as an angular distance on the circumference of the head wheel; and the diameter of the head wheel should be such that a normal bucket spacing should not cover too small an arc. In other words, the head wheel must not be too large.

When elevators inclined at about 30° from the vertical are run at speeds one-half those of Table 35 or less buckets of standard size and shape make a satisfactory discharge if they are spaced 90° apart on the head wheel—that is, if the circumference of the head wheel is four times the bucket spacing.

Table 58 shows wheel sizes and revolutions and bucket spacings for belt speeds one-half those of Table 35. It gives also the projections of buckets proper for the given spacings. In elevators designed in this way centrifugal force equals one-fourth the force of gravity and the angle A, Fig. 293, at which the main discharge occurs is $75\frac{1}{2}°$ (cosine $=\frac{1}{4}$).

At speeds lower than those of Table 58 the influence of centrifugal force is practically zero, but the data given still hold good as to wheel sizes, bucket spacings and bucket projections.

Advantages of Inclined Elevators.—Inclined elevators are often preferred to vertical elevators for reasons other than those of pick-up and discharge at slower speed. If there is a horizontal traverse from the loading point to the discharge point of not more than one-half the height through which the material is to be lifted, setting the elevator on an incline will both elevate and convey the material without the use of a feed conveyor at the foot or a

TABLE 58.—HEAD WHEELS, SPEED, BUCKET SPACING, INCLINED ELEVATOR, CENTRIFUGAL DISCHARGE AT LOW SPEED

Diameter of Wheel, Inches	Revolutions per Minute	Belt Speed, Feet per Minute	Bucket Projection, Inches	Bucket Spacing, Inches	Diameter of Wheel, Inches	Revolutions per Minute	Belt Speed, Feet per Minute	Bucket Projection, Inches	Bucket Spacing, Inches
12	35	110	3	9.4	27	24	175	7	21.2
15	31	124	$3\frac{1}{2}$	11.8	30	23	180	8	23.5
18	28	132	4	14.1	33	22	190	9	25.9
21	27	147	5	16.5	36	21	198	10	28.2
24	25	157	6	18.8					

bearing-off conveyor at the head. When, however, the angle of incline is more than 25° from the vertical, it becomes necessary to support the loaded run and to provide space for the sag of the return run (see Table 57).

One of the merits of the inclined elevator shown in Fig. 257 is that the fine damp sand which discharges late, or which shakes off the buckets at the head, does not fall down on the foot pulley as it would if the elevator were vertical. The vertical back wall of the casing permits the spill to fall straight down into a clearance space in back of the belt, from which it is picked up by the buckets when the accumulation becomes large enough. Keeping the sand from falling onto the pulley lessens the injury to the pulley side of the belt due to sand and grit between it and the pulley (see p. 290).

When a vertical elevator with centrifugal discharge is shut down in an emergency while loaded, the speed falls off, the buckets make a bad discharge and the material misses the chute and falls down into the boot, with the possibility that the boot may be choked when the elevator starts again. In an inclined elevator with the chute partly under the head wheel most of the spill in such a case will be caught and will not choke the boot.

In handling mineral pulps and slimes an elevator slightly off the vertical is preferable for the reasons stated on page 306: the spill in passing from the foot wheel to the straight run is less; the buckets hold more on the incline; the spill is less on meeting the head wheel; and if the incline is enough to permit the head chute or receiver to be placed partly under the path of the descending buckets some of the solids which are discharged late will be caught instead of spilling down into the casing. The inclination is generally limited to 10° or 15° from the vertical; otherwise, the up run must be carried on idlers, and they are troublesome in a wet elevator.

A general advantage of a belt elevator inclined at 20° or more from the vertical is that the sag between head and foot keeps the belt in contact with the foot pulley in spite of occasional neglect of the take-ups. If the foot pulley in a vertical elevator is not set down to follow the stretch of the belt the contact between them may be so slight that the pulley and shaft will not turn, and then the belt will rub and wear. If the slack is so great

that the belt is loose on the pulley the buckets are not backed up by the pulley; they do not fill well, and the capacity of the elevator falls off.

The length of the return belt is also greater than in a vertical elevator of the same height; and when the take-ups are set down to apply added driving tension to the belt, the belt can work longer and stretch more before the effect of the take-up tension is lost.

This may be expressed in a different way by saying that if the inner curve in Fig. 280 represents the hang of the belt when pulled to its maximum operating tension, and if the catenary shows the hang of the belt under no take-up tension, then the difference in length between the two is the amount the belt may stretch in service before it loses all the effect of the take-up tension. This difference may be a foot or more in an inclined elevator, but the corresponding difference in a vertical elevator may be only a few inches. When the vertical belt stretches a few inches in service it may slip on the head pulley and act badly at the foot unless the take-ups are set down; but an inclined belt may stretch several times as much before it is necessary to adjust the take-ups again.

For the reasons just stated, inclined belts do not have to be shortened and respliced so frequently as vertical belts, and a new inclined belt will run longer before it has to be cut and shortened.

In stone and rock elevators inclined at 20° or 25° from the vertical the sag of belt acts to some extent as a relief if a lump catches between the belt and the foot pulley. In a vertical elevator a tight belt is not likely to stretch still further and prevent injury when an accident of this kind occurs. In an inclined elevator, however, with some free sag at the bottom of the down belt, there may be " give " enough to prevent the stone from punching a hole in the belt. This is merely an incidental reason for inclining a stone elevator with continuous buckets on a belt; the main reason is that the incline permits the buckets to be loaded from a chute without spill (see p. 241).

Disadvantages of Inclined Elevators.—When an inclined elevator is high, or is set at a considerable angle from the vertical, it is necessary to support the loaded run to keep it from flapping and spilling material from the buckets. The idlers used for this purpose are flat-face pulleys like those made for the return run of belt conveyors; sometimes in heavy elevators with continuous buckets they are pipe rolls set every 6, 8 or 10 feet. Bearings for idlers on an inclined elevator are often so located that it is hard to inspect and oil them, and they are likely to suffer from neglect. Light elevators with spaced buckets can be put up without idlers if the slope is not far off the vertical, but heavy belts with continuous buckets generally require them.

A drawback to the use of inclined elevators is the greater amount of floor space required and the larger and more expensive casing necessary to enclose them.

CHAPTER XXIII

ELEVATOR CASINGS

An elevator casing serves several purposes: (1) to confine dust and catch material which spills at the head, and direct it to the boot where it can be picked up again; (2) to act as a guard around the moving belt and buckets; (3) occasionally, to act as a support for the head machinery.

Fig. 286.—Single Leg Steel Casing.

Wood casings are generally cheaper than steel casings, but in many places they are barred on account of the risk of fire. They are, however, preferred for wet elevators for mineral pulps and ores. The grit carried by the water that continually splashes and drips from the buckets causes destructive wear in thin metal casings, but plank resists it better, and when repairs are necessary, they can be made quickly and cheaply. For a form of such a wooden casing, see Fig. 257.

Steel casings are made in several styles. The single leg (Fig. 286) is generally used for chain elevators, but not often in belt elevators except in small sizes. For belt elevators, the double leg (Fig. 287) is more common; the back sheet of each leg is quite close to the back of the belt; it guides it and prevents it from flapping or swaying. Casings with round legs (Fig. 288) have been used in Europe and to some extent in this country. When the projection of the bucket is at least half the width of the belt the circumference of an enclosing circle is noticeably less than the periphery of an enclosing rectangle and the weight of the round leg is less; but when the buckets are relatively short in projection as compared with the width of the belt there is not much difference between the two shapes as to weight.

The round leg requires only two lines of punching and one line of riveting while a rectangular leg requires at least twice as much; but the latter has the advantage of allowing easy access to the belt and buckets by the removal of a sheet, and the sections of the rectangular casing are easily and cheaply joined by straight angles instead of forged angle rings or cast-iron flanges.

Fig. 287.—Double Leg Steel Casing.

Fig. 288.—Steel Casing with Pipe Legs.

The simplest rectangular casing for a double-leg elevator is shown at 4 in Fig. 289. It is cheap, but unless the plates are carefully squared and accurately bent in the shop the assembled sections will come crooked or with a longitudinal twist which is troublesome in erection. Casings for wide buckets are generally made of four pieces with the narrow end-plates riveted

to angles, or crimped or flanged as shown at 1, 2, 3. Casings with angle corners are easier to make dust-tight than those with flanged corners, and they are more likely to be straight and free from twist.

Dust-tight Casings.—Casings of light sheet steel, No. 14 gauge and less, cannot be made perfectly dust-tight by riveting or bolting, because the

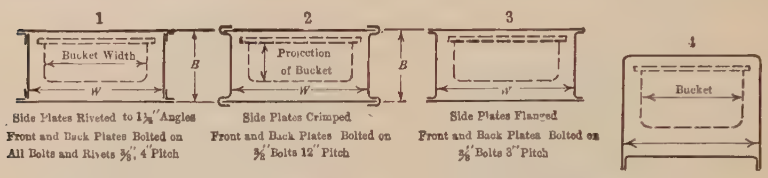


FIG. 289.—Cross-sections of Steel Elevator Legs.

edges of the sheets are stretched slightly in punching and pucker slightly between the rivet or bolt holes. When light casings must be made so tight that dust will not leak out at the seams nor at the small crevices and the corners of plates and angles, then the joints must be made with gasket strips, or the surfaces in contact may be coated with thick paint before the parts are riveted or bolted together.

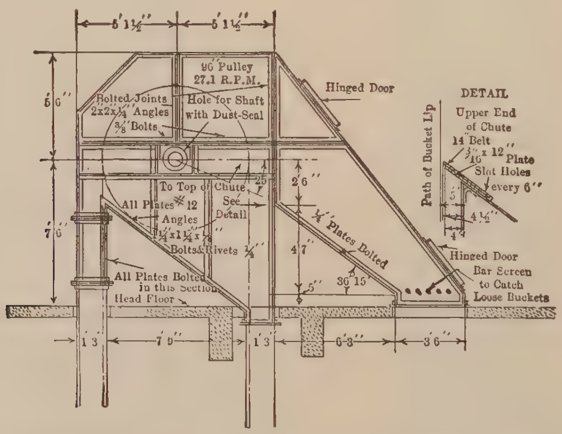


FIG. 290.—Head of Shipping Elevator, Public Grain Elevator, New Orleans. (Ford, Bacon & Davis, Engineers).

Doors in Casings.—In single-leg casings it is often not convenient nor, considering the stability of the casing, is it safe to remove an entire side plate to give access to both sides of a run of belt as is necessary for inspection and for removing or attaching buckets. Such casings should be provided with doors on each side at the levels of various floors or where it is convenient to reach the interior of the casing. If the doors are bolted the bolts should be entirely on the outside of the casing, so that they will not be lost inside when the nuts are removed.

Casings for Elevator Heads.—In the design of the head of the casing it is important to place the lip of the discharge chute low enough to catch

Fig. 291.—Man Elevator used in Flour Mills and Grain Elevators.

the discharge from the buckets at all times. On the conditions which are at times unfavorable to a clean discharge, see pages 302, 303.

Fig. 290 shows the head of a large grain elevator of recent design with

the lip placed lower than has been customary. The angle from the center of the head shaft down to the upper edge of the rubber belt which forms the lip (see detail) is 25°. All the joints at and above the level of the head shaft are bolted, as is the sloping plate which joins the two legs below the head pulley. The figure shows the bar screen referred to on page 261 and the door which gives access to it.

Other Forms of Belt Elevators.—Cleats or projections are sometimes fastened to belts to convey small boxes, packaged goods, bricks, etc., in inclined conveyors where the angle of slope is more than 20° or 30°.

Some large vertical elevators for boxes and barrels have been built with rigid or tipping arms attached to rubber or canvas belts, but they suffer from the drawback that the arms for heavy work cannot be bolted to a belt so securely, nor is the guiding of the loaded run so easily arranged as when the arms are fastened to chains.

Belt elevators with shelves (Fig. 291) are used to carry men up and down from floor to floor in grain elevators, flour mills, etc. The belt is usually 12 inches wide, 4 or 5 ply thick and runs over 20-inch head and foot pulleys at a speed of 60 to 80 feet per minute.

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